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MODERN EXAMINATIONS

OF

STEAM ENGINEERS,

OR

Practical Theory Explained and Illustrated.

WRITTEN FOR ENGINEERS BY AN ENGINEER.

COMPRISING FULL AND COMPLETE ANSWERS TO 300 QUESTIONS FOR THE USE OF ENGINEERS AND FIREMEN, WHEN PREPARING TO MAKE APPLICATION FOR EXAMINATION FOR U. S. GOVERNMENT AND STATE LICENSE; AND FOR THE INFORMATION OF ENGINE BUILDERS, BOILER MAKERS, MACHINISTS, ETC.

BY

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PREFACE.

The author takes pleasure in presenting to the steam engineers of America, a work treating in a plain, practical way of a great variety of subjects pertaining to the engine and boiler rooms, containing valuable and important information for those who aspire to become competent engineers, and something more than starters and stoppers.

The authorities quoted are reliable and up to date, and the formulas and rules given are fully explained, thus enabling all to understand and apply them.

The Manufacturers Gazette, of Boston, Mass., originally published this work, one part appearing in each weekly issue for an entire year, and as it was very favorably received by steam and mechanical engineers, boiler makers, machinists, draughtsmen, etc., it is now published in book form for the convenience of those who wish to use it in daily practice.

That the benefits derived from it by the craft may be commeasurate with the labor of preparing this volume, is the sincere wish of

THE AUTHOR.

June 22nd, 1895.

DEDICATION.

To those true friends among the engineers of the United States of America who have ever been ready to give encouragement and support, to the efforts of the author for the elevation of the position of the practical engineer, this book is respectfully dedicated.

W. H. WAKEMAN.

New Haven, Conn., U. S. A.

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MODERN EXAMINATIONS
OF
STEAM ENGINEERS.

BY

W. H. WAKEMAN.

CHAPTER I.

INTRODUCTION.—THEORY WILL DO PRACTICE NO HARM.

This book is intended to give information to those engineers who wish to qualify themselves to pass examinations and take out licenses to run stationary engines in those localities where such licenses are required. To many engineers, who are interested in the business by which they gain a livelihood, but who are employed in localities where no licenses are required, the idea of going before a board of examiners, passing an examination and receiving a legal document with a large seal thereon affixed, stating that the owner is qualified to care for and manage steam engines and boilers, is a very fascinating one, and the desire for knowledge which will enable them to pass an exammination almost universal.

The writer has been amused in noting the way in which some of them expect to get the necessary

information, for while they have a fair situation, and no license needed, they are content to merely wish that they possessed the qualifications called for in other places, and are too "shiftless" to study up the points, but when they secure a situation where their knowledge of steam engineering is to be put to a test they are very active. They will come to you full of zeal for the new conditions under which they are to work, state that they want to get posted, and give you the idea that they expect you to attach a hose to their brain, in some mysterious manner, and start up a pump, and pump them chock full of information in about half an hour. Some of them expect to get a list of questions that the inspector will ask them, together with the answers to the same, and having secured these, and learned them in the same way that a parrot or jackdaw learns to swear, that they can then go up to the pompous looking official who presides over the inspectors' department, and compel him to give one of the coveted prizes.

Reader, did you ever attend school in your boyhood days, where long lessons in spelling were allotted you, and becoming desperate on account of repeated failures, adopted a plan which you thought would enable you to go home when the other boys did? This plan was to note your position in the class (near the foot), and select the words which it would naturally be your turn to spell, and learn them thoroughly. But do you remember how you were seized with consternation when you discovered that the teacher began at the other end of the class to give out the words, thus putting those to you that you had not even attempted to learn to spell? If you have been through this experience, you were probably

impressed with the idea that the only sure way was to learn the whole lesson. And so it will be with men who expect to learn a few questions and answers, and so get a first-class license, for they will find that the answers that they did not learn are just the ones that they will be required to give, and consequently the only sure way, if a first-class license is needed, is to learn the whole lesson. The whole lesson did we say? That means a great deal, for probably none of us have accomplished that feat, and in this respect the comparison does not hold good, but at the same time, if all engineers who carefully read the whole of this series of articles will make themselves familiar with all the subjects treated of, together with those that may be suggested by them, they will not fear to go before any board of examiners, and submit themselves to a fair test as to their knowledge of engines, boilers, etc.

Occasionally we meet a man who claims that he knows how to run an engine, but that he cannot tell others how it is done, but such an excuse appears to be very unsatisfactory, for while we may not be able to express ourselves in as correct language as others do, still if we really know how these things are done, we can usually find some way to tell it to others. The fact is that there are many good men in charge of steam plants who were formerly firemen, and through faithfulness and sobriety have been promoted, and get along after a fashion, without really knowing anything of the theoretical part of the business. Such men can keep their plants clean, practice economy in the use of supplies, and are devoted to the interests of their employers, but at the same time just as soon as any question comes up which requires a little "head-work" to settle, they are

ever ready to call upon others to help them out of their troubles, although at other times they affect to despise them, on account of being what they are pleased to term "book engineers" or "paper engineers." Well, we are of the opinion that the well-posted ones can stand it if the others can, for it is a well-known fact that these men speak disparagingly of their fellows to divert attention from their own defects.

I trust none of my readers are ready to admit that they are in this class, but if there are any such, allow me to ask a few questions. As you now claim to understand the practical part of your business, do you think that it would detract from the knowledge already possessed if you should acquaint yourself with the theoretical part of your occupation? As you now know that it is well to have your safety valve weight set so that the valve will open when the gage indicates 80 pounds pressure, and find it a good plan to lift the valve from its seat once every day, would it do you any harm if you knew how to determine the area of said valve, and also how to calculate the pressure at which it should lift, all necessary data being given, and then compare it with the results obtained in practice? Are you aware that such knowledge will tend to lighten the work in the fire room, and relieve the monotony of the ceaseless routine of the engine room? We beg pardon for this digression, and will return to our subject. While a list of direct questions are necessary in examining a candidate, still it is not the intention of the writer to give a catechism at this time, but the subjects will be treated in a more general way, believing it to be the best plan.

CHAPTER II.

GENERAL OUTLINE OF SUBJECTS THAT SHOULD BE FAMILIAR TO APPLICANTS FOR OTHER THAN FIRST CLASS LICENSES.

Applicants for a license other than first class are expected to have a general knowledge of the plant which they are to have charge of, for in these cases the license is given to run a certain plant, and will not qualify a man to take charge of any plant that he may be hired to run. A misunderstanding concerning this point has caused some rather embarrassing mistakes, for if a man conceives the idea that it would be a help to him in securing employment, provided he has a license to show, and, acting on the happy thought, applies for one on general principles, about the first question that he will be asked will prove to be a hard one to reply to, for he will be asked to state where he is to be employed, and as this is the very question that is puzzling him in his own mind, he can scarcely be expected to give an intelligent reply. But having secured a situation, his next care should be to get a good idea as to the details of his plant. He should know the diameter of the cylinder of his engine, the length of the stroke and the number of revolutions per minute. He should be able to state whether it is an automatic or an adjustable cut-off, or one of the fixed, cut-off, throttling type; the size of the steam and exhaust pipes should be given, and a

general description of the whole engine prepared, so that almost any question as to the size of any of the parts can be answered. The boiler or boilers should also receive attention, and a general description of them be prepared, including their diameter and length, the thickness of the material of which they are composed, and whether they are of iron or steel, the number and the diameter of the tubes or flues in them, the size of the furnaces and the kind of setting adopted. The boiler feeders should not be overlooked, but the diameter of the water pistons or plungers given and the length of their stroke. Whether an injector is provided or not, and, if so, is it of the lifting or the non-lifting type, and its rated capacity should be known; this to be determined by reading the maker's catalogue, and an opinion given as to whether the capacity of each of them, in cases where more than one boiler feeder is provided, is sufficient for the needs of the plant.

It will do no harm for the candidate to review some ordinary questions concerning his everyday practice in the engine and boiler room, and as to what he would do in cases of emergency. He should remember that his first duty on entering his boiler room in the morning is to ascertain where the water level in the boiler is, this always to precede the opening of the damper. It is not sufficient to know that there was three gages there the night before, and that, as the floor is not flooded, it demonstrates that it is still there, for there are several ways whereby it might have made its escape unnoticed.

Probably there are few engineers who have not at one time or another found less water in the boiler than they anticipated, and although month

after month slips away and the above-mentioned precaution seems entirely unnecessary, still it should never be neglected, for no one can tell when such neglect will cause serious damage to be done. This also should receive all due attention before firing up a boiler after it has been cleaned.

He may be asked what he would do in case that he accidentally allowed the water to get low while the engine is running, to which he should always reply that he would never allow such a thing to happen, and if it is suggested that even the best of engineers forget to attend to their duties at times, he should never admit that it is possible for him to forget so important a thing as that. If, however, the feed pipe should burst, thus allowing the water to escape faster than it can be replaced, he should run the pump to its full capacity, and, provided he can open the furnace door without being scalded, he should cover the fire with fresh coal or damp ashes, for the writer believes that this is a much better plan than to attempt to draw it, as when a fire is disturbed, it gives out a very intense heat for a few minutes. Of course a pipe may burst in such a way as to fill the furnace with steam and scalding water, thus preventing the fireman or engineer from banking it, but in that case it will be so deadened down that it will not be capable of injuring the boiler.

The size of the safety valve should be noted, and an opinion formed as to whether it is large enough for the purpose intended or not, and it is well to bear in mind the fact that careful engineers try their safety valves at short intervals, in order to know that they are ready to open when needed.

It is the duty of an engineer to bank his fire

and shut down the plant whenever he is convinced that the boiler has developed a weakness, caused by a blister, a fractured plate, or any of the other agencies which work to cause its destruction, when such weakness has advanced sufficiently to warrant it, and no consideration should cause him to continue to run with a high pressure, thus endangering lives and property.

The applicant should familiarize himself with the construction and operation of the slide valve, as it is quite possible that he may be shown a model and asked to set it properly, and if he should find that it is so made that it cannot be set so as to work properly, it would not be the first time that an engineer has been so tested.

These questions, as will be noted, are of a plain, practical nature, and every man who claims to be an engineer should be able to readily answer them.

CHAPTER III.

DIRECTIONS TO THOSE WHO DESIRE A FIRST CLASS LICENSE.

We now come to matters with which the candidate for the honor of holding a first-class license must be familiar, in order to be sure of passing a satisfactory examination. But a small proportion of the licenses issued are for this class, because it is much easier to get one of the lower grade, which usually answers every purpose.

The applicant is deterred from seeking one by the fact that if he applies for one of a lower grade (but which may be sufficient to answer his purpose), if he fails to pass the examination, he may try it again in a few days, but if he wants a first-class license, more will be naturally expected of him, and furthermore, if he cannot meet the requirements at his first trial, he may be obliged to wait for three months before another trial will be granted him, and in the meantime the situation that he wished to take will be secured by someone who has already passed the ordeal once, and consequently knows just what to expect. We would not be surprised if the examination were far less rigid after a man has become well known to the inspector.

It is perfectly natural in speaking of the different parts which constitute a steam plant to mention the engine first, which rule will be followed here, and in enumerating the parts of the engine

we usually speak of the valve, and the machinery which operates it, first, and therefore we would say that it is well for the applicant for a license to be thoroughly familiar with the principles here involved. There is a great difference in the ways that men go about learning these things, for some of them appear to know no other way than to learn just how the several parts of a certain valve gear are to be adjusted in relation to each other, without understanding any of the principles which lie at the foundation of their operation. For instance, a man of this stamp will take charge of an engine, and among the first things that he will do will be to note the position of the eccentric on the crank shaft, the exact length of the eccentric rod, and also of all of the other parts which are adjustable, and mark them in such a way that he will be able to replace them, should they ever become disarranged from any cause.

Nearly all valves if properly made have lap, and if properly set they will haye lead. This statement naturally suggests the question, what is lap? Let us suppose that the steam ports, or passages leading from the steam chest into the cylinder, are 1½ inches wide, and when the slide valve is placed in the centre of its travel, the ends of it being made to cover up these ports, each end of it laps over the ports one quarter of an inch. That part which laps over the edge of the port towards the end of the valve is called outside lap, and that part which projects over the part toward the inside or middle of the valve is called inside lap. These may not be good proportions but will do for illustration.

The object in putting outside lap on a slide valve is to save steam by cutting off the supply from the boiler before the piston reaches the end

of the stroke. The object sought in putting on inside lap is to close the exhaust port before the piston reaches the end of the return stroke, thus trapping in some of the exhaust steam, (filling the clearance), and raising it to some considerable pressure, according to the speed of the engine and the judgment of the engineer. The effect is to make the engine run quietly by bringing the reciprocating parts to rest, by the piston striking against a cushion, as it were, and also to save steam, for the portion of the wasteroom at the ends of the cylinder, which are thus filled with exhaust steam will not have to be filled with live steam..

The term lead, as applied to the working of the steam engine, is partially self-explanatory, for it means that the action of the valve in opening must lead the action of the crank, taking the dead centre as a starting point; therefore if when the engine is on one of its centres the valve is open one-sixteenth of an inch, we say that it has one-sixteenth of an inch lead. Of course we mean that it must be open on the same end that the piston stands at, and care must be taken to see that the eccentric is set so that when the engine is turned over in the direction that it is to run, the valve will continue to open, for it is possible to set it so that while it may give the valve lead, still if the engine be turned over by hand, the valve will immediately close, and it is quite plain that it will not run even if steam is admitted under those conditions.

These principles apply to all valves of whatever form, size or description, and it makes no difference whether they slide or rotate, lift or swing to give opening of port. It is the same whether the steam is admitted over the end of the valve or

through the middle of it. Whatever devices are used to turn the rotary motion of the crank shaft into the reciprocating motion of the valves does not affect them, and if engineers would be masters of this idea it would save much hard work in studying out how to set different valves. I do not mean to say that the eccentric would be in the same position on the shaft in all of the cases, for it would not, but the motion which opens the valve must precede the motion of the crank in every case.

The connections between the crank shaft and the valves are made in different ways, and this will effect the position of the eccentric, for if a rocker arm is hung at the bottom the eccentric must be put in a certain position to do its work properly, but if the rocker arm is changed and hung in the middle, the eccentric will have to be put in an entirely different place, but the principle is the same, for the motion of the valve must be in advance of the crank. To explain just what to do to set the valves of every kind of engine that is built would fill the pages of a large book, but if the engineer understands these principles, and then will carefully examine the engine that he is to adjust, he will soon see how the desired result is to be obtained. To reverse the engine, put it on the centre, and note the amount of lead that the valve has. Loosen the set screws, and turn the eccentric in the opposite direction from that in which it is intended that the engine shall run, until the valve has the same lead on the same end. Tighten the set screws, and the valve will be properly set, assuming that it was so before.

CHAPTER IV.

REVERSING AN ENGINE.—THROW OF THE ECCENTRIC.—STEAM AND EXHAUST PIPES.

While on the subject of reversing an engine, it is proper to speak of rules given by others for this purpose, which do not agree with the one just given, for when the seeker after knowledge meets with two or more rules which do not agree, he is perplexed to know which one to believe. In two different books which give a so-called rule for doing this, one of which claims to be the only practical work published, I find the following instructions given: To reverse an engine, place it on the dead centre, remove the steam chest cover, and note the amount of lead that the valve has. Loosen the set screws in the eccentric, and turn it in the opposite direction until the valve has the same lead on the opposite port; then the engine will run in the opposite direction. This is a mistake, for when an engine is to be started from the dead centre (or a little past the centre) it must take steam on the same end, regardless of the direction in which it is to run. If an engine is on one of its centres, and the valve admits steam to the opposite end of the cylinder from where the piston stands, how can it be started in either direction? In the first place, the whole volume of the cylinder must be filled with steam, which would constitute a very large clearance, to say the least, and then as the piston is already at

the end of the cylinder, how can it be forced any further in that direction?

In turning an engine over from one centre to the other, it is interesting to note the difference in the travel of the crank pin and the wrist pin. There is before me, as I write the explanation of this problem, in a book written for the purpose of instructing engineers in their duties: "A crank pin travels 1.1416 times further than the piston each revolution, or 0.5707 times further each stroke. For example, take an engine with a 12-inch stroke, the piston travels 24 inches and the crank pin 37.6992 inches each revolution, or the piston travels 12 inches each stroke and the crank pin 13.6992 per single stroke of the piston." This statement is about as badly mixed up as it could be, for in the case of an engine with a 12-inch stroke, the crank would be 6 inches long, and in making a revolution it would describe a circle whose diameter is 12 inches, and whose circumference is 37.6992 inches, and in making a single stroke or half a revolution, it will travel through just one-half of that distance.

A few points about the eccentric may be of interest at this point. An eccentric is to all intents and purposes a crank and a crank pin. The distance from the centre of the eccentric to the centre of the hole bored in it for the shaft represents the length of the crank, and twice this distance gives us the stroke or throw of the eccentric, just the same as twice the length of the crank of an engine gives us the stroke of it. It is seldom convenient to take eccentric off from its shaft and measure these distances, therefore if it is desired to find the throw of it, while in position, measure the distance from the shaft to the edge of eccentric on the heavy side, and also on

light side. Subtract one from the other and the remainder will be the throw of the eccentric.

The diameter of the eccentric represents the diameter of the crank pin in this case, and although these proportions are very different from those used for the crank of an engine, still the principle is exactly the same. Sometimes engineers will ask this question: "If an eccentric is turned down, say from 10 inches in diameter to 9 inches, will it affect the travel of the valve?" We wish to ask a question. Suppose that we take an ordinary crank on an engine, and this crank projects beyond the crank pin to the extent of 3 inches. If we turn it off so that it is 1 inch shorter than it was before, without disturbing the crank pin, will it affect the stroke of the engine? Of all those who read this chapter, probably not one will claim that it would, because as we have not changed the distance between the centre of the crank shaft and the centre of the crank pin, and this distance is what determines the stroke of the engine, so in putting the eccentric in a lathe and turning it down an inch, we have not changed the throw of it, because we have not changed the distance between the centre of the eccentric and the centre of the hole for the shaft.

The applicant for a license should be able to calculate the size of steam pipe needed for any engine, and the old-time rule for this is that it should be one-quarter the diameter of the cylinder. This may do very well for the slow piston speeds formerly used, but if that is sufficient for a speed of 300 feet per minute it does not necessarily follow that it will give the best results for a speed of 600 feet. There is no unqualified rule that will apply to all cases, for

it is quite evident that an automatic engine will need a larger steam pipe than a throttling one of the same diameter will, for in the former case all the steam must be admitted before the cut-off valve closes, which may average at quarter stroke, while in the latter case, with a cut-off fixed at three-quarter stroke, the pipe will be admitting steam during a space three times as long as in the former to do the same work.

For an automatic engine running at a high piston speed, the pipe should be one-third diameter of the cylinder, but for a throttling one, if it is one-quarter the diameter of the cylinder, it will answer every purpose, for there is such a thing as having a pipe too large, causing greater loss from condensation. If the exhaust pipe is four-tenths of the diameter of the cylinder, it will be large enough to allow the steam to escape freely, without causing back pressure. A smaller pipe will answer every purpose if the engine has only a light load, but when this is increased to the full capacity of the machine, unless it is of ample size, it will cause unnecessary back pressure, and that at a time when it will be of the greatest disadvantage.

CHAPTER V.

STEAM AND EXHAUST PORTS.—DIMENSIONS OF DIFFERENT PARTS OF AN ENGINE.

The size of the steam and exhaust ports should receive due attention, and as the practice of the prominent engine builders in designing them does not agree, it is rather puzzling to any one who may seek to show that it will not do to depart from a rule that he may have adopted. One firm who manufacture a great many engines make the steam ports six per cent. of the area of the cylinder and the exhaust ports 12 per cent. of it, while another high-grade engine builder makes them 8.7 per cent. and 10 per cent. respectively. These are good samples of their class, and from the results obtained from them it is safe to say that the steam ports should have 10 per cent. of the area of the cylinder, and the exhaust ports 12 per cent. of the same.

The reasons for this conclusion are as follows: The engines having a steam port area of six per cent., do not maintain the initial pressure to the point of cut-off while running at a piston speed of 500 feet, but the way that they exhaust the steam, even when very heavily loaded, is very satisfactory, while the engines with the 8.7 per cent. steam ports hold the steam line up remarkably well, although they are run at a slow speed, while an improvement could be made in the exhaust ports, hence the conclusion that the steam port area

may be made 10 per cent. of the cylinder area, and exhaust port area 12 per cent. of it, with good results.

The diameter of the crank shaft should be one-half of the diameter of the cylinder, in order to be stiff enough to avoid springing, with its accompanying evils. Of course it is easy to find engines that have them smaller than this, and still give no trouble, but on the other hand, those builders who take pride in showing that they put plenty of stock into their product adopt the above rule for size of crank shaft.

If a deep key seat is cut in this shaft it weakens it materially, and to avoid this, at least one prominent maker (and perhaps more) has the shaft made larger where the fly wheel is to be located, so that the keyway does not make it weaker than the main part of the shaft. If the diameter of the crank pin is .25 of that of the cylinder, it will give good results.

As the wrist pin is supported at both ends, it is not necessary that it should be as large as the crank pin, so far as strength is concerned, but to make the wear on these pins as nearly equal as possible, it is frequently the custom to make them of the same diameter or nearly so.

The connecting rod should be .23 of the diameter of cylinder in the middle and .18 of it in its smallest parts, and in order to reduce what is usually called the "angularity of the connecting rod" as much as possible, its length should be six times the length of the crank.

Sometimes we find a connecting rod made solid at one end and with the ordinary strap and key at the other end. This is done to keep the length of the rod, or, more properly speaking, the distance between the centre of the crank pin and the

centre of the wrist pin constant, for with the strap and key the wear tends to shorten this distance, and with the solid end it tends to lengthen it, so that it will always be the same, provided the brasses wear alike.

The diameter of the piston rod should also be determined by the diameter of the cylinder, with perhaps some modification for very high steam pressures. With ordinary lengths in the case of a non-condensing engine it should be made .15 of the diameter of the cylinder, for say 90 pounds or less, but if 130 pounds are to be required, .18 of it will be necessary for safety.

It is the custom of some of the builders of medium-sized machines to make the length of the main bearing the same as the diameter of the cylinder. This is a very liberal allowance, still it is not necessary or practicable in the case of large engines.

The lengths of the eccentric and valve rods are determined by the size of the engine, and their diameters vary greatly, and always will, for with a style of valve that works easy a light rod will answer every purpose, but with the same size of engine having a different valve it may require rods nearly twice as large as in the former case. When we speak of a counterbore in a cylinder we mean that it is bored larger at each end for a certain distance, in order that the packing rings, as they slide back and forth in the cylinder, may not leave a shoulder at each end of it, which in the course of time would cause a pound when the engine passes its centres.

As the rings pass partially over the edge of this counterbore, it is plain that a shoulder cannot be left there until the main part of the cylinder is

worn to the full size of it, but by that time the cylinder will need reboring.

Engine builders frequently make the mistake of not counterboring deep enough to allow the rings to travel to it, which can sometimes be remedied by putting in wider rings.

CHAPTER VI.

CONDENSING AND NON-CONDENSING ENGINES.—DESCRIPTION OF DIFFERENT TYPES OF ENGINES.

It will probably be necessary to explain the difference between non-condensing and condensing engines, by stating that with the former steam is exhausted against the atmosphere, while in the case of the latter it is exhausted into a condenser, and the air pump removes the pressure of the air, the effect of which is to reduce the back pressure. This will naturally suggest the question as to what the back pressure on an engine is, to which the engineer should reply that the steam acting on the piston to drive it forward gives us the forward pressure, and as there is always some pressure on the opposite side of the piston which tends to oppose its advance, this is very properly called back pressure, and the difference between these two is called the mean effective pressure.

In giving the amount of these different pressures it is customary to state what the average is for the whole stroke, or in other words what the mean pressure is. When the term back pressure is used, it is generally understood to mean the back pressure above the pressure of the atmosphere, as for instance when we say that our engine is working under four pounds back pressure, caused by heating the factory with exhaust steam, we mean that there is four pounds back

pressure above the atmosphere, but when mention is made of the back pressure absolute, we mean the total back pressure measured from a perfect vacuum. By initial pressure is meant the pressure in the cylinder at the beginning of the stroke, and by terminal pressure is meant the pressure existing at the time that the exhaust valve opens to discharge the steam.

The applicant for a license may be called upon to explain the difference between a throttling and an automatic engine, when he should reply that with a throttling engine the point of cut-off is fixed, and the speed regulated by a valve in the steam pipe, called the governor valve, which admits more steam when the load is increased and less when it is decreased, but during a fixed portion of each stroke (usually about three-quarters of it), while with an automatic engine the steam is admitted at nearly boiler pressure, and when sufficient has been admitted to complete the stroke it is quickly cut off.

An adjustable cut-off engine is one in which the speed is regulated by throttling the steam, and the point of cut-off is varied by means of a hand wheel on the outside of cylinder.

A compound engine is one in which the steam is used in one cylinder, then exhausted into another, where it does more work, and is then exhausted into the atmosphere. A compound condensing engine is a compound engine with the condenser attached.

A triple expansion engine is one in which the steam is used in one cylinder (called the high pressure cylinder) then exhausted into another called the intermediate, and then exhausted into another called the low pressure cylinder, and from thence into the atmosphere. If a condenser

is attached it is said to be a triple expansion condensing engine.

A quadruple expansion engine is one in which four cylinders are used, the same steam passing through all of them, and then into the air, and if a condenser is added, it is called a quadruple expansion condensing engine.

The advantages and disadvantages of these several types may be summed up as follows: The throttling engine is used extensively for small mills and factories, on account of its simplicity and low first cost, but it is very wasteful of steam, and could not be tolerated for large powers where fuel is worth anything. The automatic engine is usually more complicated and consequently more liable to become deranged than the throttling engine, but is more economical in the use of steam, and for medium powers is very satisfactory. When a condenser is added it increases the complication, but reduces the consumption of fuel.

An adjustable cut-off engine is a kind of a cross between an automatic and a throttling engine, but can be used to advantage where the load is reasonably constant during a part of the day, and is increased to a certain known extent during the remainder of it, for in that case the cut-off may be varied to suit the load. It is not probable that they will ever come into very general use for stationary work.

The compound engine is economical in the use of steam only when the load is just right for it, but is expensive to run if the load is too light, and cannot be used to advantage if the load is a very heavy one. It is also complicated, and the first cost of it is large.

The compound condensing engine is coming

into very general use for large powers, on account of its economy in the use of steam, and although it is complicated, and the first cost of it is very large, and intelligent supervision is absolutely necessary, still its economy more than overbalances these objections to its use. A high boiler pressure is essential to success.

In the triple expansion condensing engine, the complication and first cost is increased, but owing to its greater economy it finds favor with some. The first cost of the quadruple expansion condensing engine is excessive, and the cost of keeping it in repair proportionately great. It requires the most intelligent supervision that can be procured, and its use is of necessity limited to the very largest powers, but where fuel is expensive it is a profitable investment. A very high boiler pressure is absolutely necessary in order to use the steam to good advantage.

We sometimes wonder what sort of a machine the steam engineer of the future will be called upon to run. Will it have six, eight or ten cylinders? Just imagine an engine room containing an engine with ten cylinders, each one of which has four valves, with a total of 20 dash pots. It might do very well as long as everything is new and in good order, but how will it be when it has run for several years, with a record of only $1.37 spent for repairs, and the boss positively declining to make any that year, because business is dull? And then, as a matter of course, the boiler pressure will have to be increased to a frightful extent, and boiler plates will compare favorably with the armor plates for the new cruisers that will be in course of construction at that time. But some will say that we have now reached the limit for all these things, but do not be too sure

of it, for you must remember that there were men who believed at one time that an engine could never be run at 150 revolutions per minute.

CHAPTER VII.

MEANING OF THE TERM "HORSE POWER."

The term "horse power" is one that is very commonly used, but at the same time there are many engineers, or men in charge of steam plants, that do not thoroughly understand it.

Let us take for our starting point a small engine made in the usual way, except that instead of a fly wheel it has a drum on the crank shaft, on which is wound a small wire rope. We have a piece of cast iron which weighs 33,000 pounds. We attach the end of the rope to the weight, and start up our engine, which runs at 50 revolutions per minute. We time the weight as it goes up, and find that it rises just one foot per minute, with a mean effective pressure of 20 pounds in the cylinder. This is just one horse power.

Now suppose that we attach two pieces of cast iron, each weighing 33,000 pounds, to the end of our wire rope, and start up the engine. It runs until the slack of the rope has been taken up, when it stops, and we discover that we cannot raise the two weights together. We now proceed to take the drum off from the crank shaft and put a gear wheel on in its place. We next put up a second shaft, on which we put the drum, and also a gear wheel just twice as large as the one on the crank shaft. We now start up the engine with its speed of 50 revolutions, and a mean effective pressure of 20 pounds, as before; we find that we

can just raise the two weights. Are we now using two horse power?

Some will probably say yes, because we are raising just twice as much weight as we did before, but in reality we are using just one horse power, as we did in the first experiment, for although we are raising twice the weight that we did before, it travels upward but one-half as fast, therefore the power consumed is the same in both cases.

Now if we speed our engine up to 100 revolutions per minute with the same pressure as before, our weights will travel upwards at the rate of one foot per minute, and then we shall be developing two horse power. If we increase the speed to 200 revolutions per minute with the same pressure, we shall raise the two weights at the rate of two feet per minute, and shall use four horse power to do it.

We assume that the highest point of speed has been attained, and we still wish to do more work. Our next move is to increase our mean effective pressure to 40 pounds, and then we can raise four pieces of cast iron, each weighing 33,000 pounds, at the rate of two feet per minute, and shall consume eight horse power in doing it.

By this explanation we trust that the meaning of the term "horse power" is made plain, for it is 33,000 pounds raised one foot high in one minute, and it very naturally follows that to determine the horse power of any engine we must ascertain the number of square inches in the surface of the piston, and multiply this number by the effective pressure acting on every square inch. This product must be multiplied by the number of feet that the piston travels per minute, and

when we divide the product so obtained by 33,000 we shall know the horse power of the engine.

We sometimes hear men say that they wonder how it is that such a small engine can do so much work, as for instance in the cases of the heavy road rollers that we see on our streets. The engines that drive them are small, but we do not always consider that while they are moving a heavy load still it travels but very slowly. If the same engine were put into a light wagon it could make it travel very swiftly, and few men that see it would consider it quite possible that it required fully as much power to drive the light wagon swiftly as it does to drive the heavy road roller slowly.

We also hear men speak of putting on a small fly-wheel, and belting on to a large pulley on the main shaft, in order to get more power out of their small engine, but with a constant piston speed and mean effective pressure, the power of the engine is the same, regardless of the way in which it is connected to the machinery to be driven.

In the foregoing illustrations the friction of the engine itself and of the gears was not spoken of, because it was unnecessary, for while it increases with speed and pressure, it has no effect on the raising of the weights, which represent the machinery in a factory.

If a man says that he is running a 100 horse power engine, in reply to a question, what an indefinite answer it is. Take the case of the first illustration. There we had an engine of one horse power. In the last illustration we had an eight horse power engine, and yet it was just the same machine in both cases. It had not been enlarged in any of its parts, or strengthened in any way, and when it is shut down it will have exactly

the same appearance in both cases, and yet one day it is a one horse power machine and the next it is eight times as large, if we were to judge by its rating. It is just so with larger engines, for the principle holds good with them, as with smaller sizes.

If an engineer is asked how large an engine he is running, and replies that it has a 20-inch piston, a 4-foot stroke, and runs at 65 revolutions per minute, with a boiler pressure of 90 pounds, he conveys an intelligent idea of the machine in his charge, and if he adds that the mean effective pressure is usually about 35 pounds, the reply is complete, but not otherwise.

CHAPTER VIII.

CALCULATING THE HORSE POWER OF AN ENGINE. —SQUARE ROOT.

The matter of figuring the horse power of an engine should be thoroughly understood by the applicant for a license, and by this we do not mean that he should simply know how to repeat the rule, but that he ought to be able to not only work out an example illustrating the workings of the rule, but if other questions are asked him on the subject he must be prepared to give a ready answer. To fully illustrate what is meant by the above, an example will be introduced here and worked out in full, so that it may be made plain. The horse power of an engine is required, the following data being given: Diameter of cylinder, 18 inches; diameter of piston rod, 2.75 inches; stroke, 42 inches; speed, 65 revolutions per minute; mean effective pressure, 35 pounds. To find the area of a piston 18 inches in diameter we multiply 18 by 18, and the product by .7854, and find our answer to be 254.46 square inches. The area of a piston rod 2.75 inches in diameter is 5.93 square inches. Dividing this by two our quotient is 2.96. Subtracting this from the full area of the piston we have 254.46−2.96=251.5 square inches, which is the effective area of the piston. The area of the rod is to be deducted because the space occupied by it is not accessible for the steam to act upon, and we subtract but one-half

of it because it is on but one side of the piston, the whole of the other side being effective. Multiplying the area of the piston by the speed, and this product by the mean effective pressure, and dividing by 33,000, we have 251.5×455×35÷33,000=121.37 horse power. This part of the problem is very easy to learn, but when it is presented in another light, or perhaps it would be better to say that if other parts of data are given, and other parts are to to be supplied by the candidate, it may prove a stumbling block to him, and undoubtedly will unless he has given the matter some thoughtful consideration. Suppose for instance, that he should be asked to state at what speed an engine should be run whose cylinder is 18 by 42 inches, with a mean effective pressure of 35 pounds, in order to develop 121.37 horse power. In that case he would multiply 121.37 by 33,000, and divide the product by the mean effective pressure, and the quotient so obtained by the area of cylinder or piston, 121.37×33,000÷35÷251.5=455 feet per minute. The piston travels seven feet per stroke and 455÷7=65 revolutions per minute.

Suppose the question asked was, what the diameter of the piston should be if the stroke were 42 inches, the speed 65 revolutions, the mean effective pressure 35 pounds, and the horse power developed 121.37? In this case he would multiply the power developed by 33,000, and divide the product by the mean effective pressure, and the quotient so obtained by the speed in feet per minute: 121.37×33,000÷35÷445=251.5. To this must be added one-half of the area of the piston rod, which is 2.96, and 251.5+2.96=254.46 square inches. Now in ascertaining the area of the piston, we multiply the diameter of it by itself

and the product by .7854, consequently our next move in solving the problem is to divide 254.46 by .7854 and the quotient is 324.

We must now find the square root of 324. The following rule for obtaining the square root of any number that can be divided by four without a remainder is inserted here, because it will answer every purpose for many of the numbers that engineers will use in making the above calculation. Divide the number by four and find the square root of the number so obtained. Multiply this by two and the product will be the desired square root. Applying this rule to the case in hand we have $324 \div 4 = 81$. This being a small number it requires only a moment's consideration to show us that the square root of it is 9, and $9 \times 2 = 18$, therefore the square root of 324 is 18. If our calculation was for a 24-inch cylinder, the number that we wish to obtain the square root of would be 576 instead of 324, and $576 \div 4 = 144$, and it needs but a moment to see that the square root of 144 is 12, and $12 \times 2 = 24$. And even if our engine were a 30-inch one, it would still be convenient to use this rule, for the number would then be 900, and $900 \div 4 = 225$, and the square root of 225 is 15, and $15 \times 2 = 30$.

There are, however, other sizes, both larger and smaller than these, which this rule will not answer for as well, and for such we can use the following rule for whole numbers: (1) Point off from right to left, in orders or places of twos. (2) Ascertain the highest root in the first order and place it at the right of the number, as in long division. (3) Square this root and subtract it from the first order. To the remainder annex the next order and double the root already found and place it to the left of this dividend. (4) As-

certain how often this divisor is contained in all but the final figure of the dividend, and place the quotient to the right of the root already obtained, and to the right of the trial divisor. (5) Multiply this divisor by the final figure in the root, and subtract as before. In like manner proceed until all the orders have been worked. In ordinary business practice it is much more convenient to take up a book of reference, which is usually at hand, and refer to the table containing the desired information, when the square root of a number is wanted; nevertheless, as this is something of value to the engineer who wishes to become well posted, and as in the writer's opinion the rule, as given in some of the text books, is not readily understood, an explanation of its workings will be given in the next chapter.

CHAPTER IX.

TO INCREASE THE POWER OF AN ENGINE.—EFFECT OF ADDING A CONDENSER.

In order to illustrate the workings of the rule for extracting square root of a number, and applying it to the case in hand, we will set down the number, which is 324, and our rule says: "First, point off from right to left in orders or places of twos." This brings our point between 3 and 2, giving us 3 in the first and 24 in the second order. "Second, ascertain the highest root in the first order," and that means to take the highest number which when multiplied by itself, the product will not exceed the number in the first order, "and place it at the right of the number, as in long division"; thus, 3.24(1. "Third, square this root," which means multiply it by itself, "and subtract it from the first order: 1×1=1 and our example is then as follows:

3.24(1
1
———
2

"to the remainder annex the next order" as follows:

3.24(1
1
———
224

"double the root already found, and place it to

the left of this dividend." The root already found is 1, and when it is doubled we have 2, and having placed it to the left of this dividend our example stands thus:

3.24(1
1
———
2)224

"Fourth, ascertain how often this divisor is contained in all but the final figure of the dividend." Now the final figure of our dividend is 4, and leaving this off we would naturally say that 2 is contained in 22, 11 times, but this is not proper here, for it can in no case exceed 9, and as we can only assume a number for this place, and try it, in order to know whether it is correct or not, let us try 9, and place the quotient to the right of the root already obtained, and to the right of the trial divisor. When we have done this our example stands thus:

3.24(19
1
———
29)224

"Fifth, multiply this divisor by the final figure and subtract as before." This divisor is 29 and the final figure in the root is 9, and $29 \times 9 = 261$. Now we cannot subtract 261 from 224, therefore we must try a lower number, and we will take 8. Our example is now as follows:

3.24(18
1
———
28)224

We now multiply 28×8 and our product is 224, which we subtract and find that we come out even, and the problem is solved:

```
    3.24(18
    1
    ———
 28)224
    224
    ———
      0
```

The square root of 324 being 18, and therefore the piston must be 18 inches in diameter.

Again, the diameter of the cylinder, the stroke, the speed and the horse power may be given, and the necessary mean effective pressure required. In this case we multiply the horse power developed by 33,000, and divide the product by the speed, and this quotient by the area of the piston, the quotient so obtained being the mean effective pressure. $121.37 \times 33{,}000 \div 455 \div 251.81 = 35$ pounds. The horse power constant of an engine is found by multiplying the effective area of the piston in square inches by its speed in feet per minute, and dividing by 33,000. The horse power may be determined by multiplying the horse power constant by the mean effective pressure, if so desired. Sometimes an engine will not do the work required of it, and it may be impracticable to increase the speed of it. In such a case if we can increase the mean effective pressure, we shall obtain more power with which to drive machinery in direct proportion to the increase excepting the increase in the friction of the engine itself. This may be done by increasing the boiler pressure, or the size of the steam pipe, and making it more direct in reaching the engine; by putting on a more suitable governor; by enlarging the steam ports, or by lengthening the point of cut-off. To determine just which of these to do requires good judgment, for it is quite plain that we shall not be

able to remedy the evil unless we know just where the trouble lies. If we get boiler pressure approximately up to our throttling governor, with a full load on the engine, we shall know that our steam pipe is large enough, and if practicable the boiler pressure should be increased.

If we get boiler pressure in our steam chest we shall know that our pipe and throttling governor are not at fault, but if there is less pressure in the cylinder than in the steam chest, either the steam ports are not large enough, or the valve is not set properly. If our slide valve cuts off the steam at half-stroke, we can increase the mean effective pressure by cutting some of the lap off from it, thus allowing the steam to follow during a greater portion of the stroke. If this is done it will be necessary to reset the valve. The question as to whether we will have boiler capacity sufficient for the new conditions must receive attention before alterations are made. Adding a condenser will make it possible to increase the mean effective pressure, but does not of itself increase the said pressure. If the mean effective pressure cannot be increased, or the engine speeded up, the power of it may be increased by putting on a larger cylinder, but of course this cannot be carried to any very great extent, for it is only fair to assume that when the engine was built the several parts were calculated for each other, with a certain margin for safety, and if we increase the size of the cylinder very much, the stress may be too great for some other part to bear, and a wreck be the result. When an engine is overloaded, the best way is to remove it, and put in a new one, adapted to the load to be carried.

CHAPTER X.

TO DETERMINE THE SPEED OF AN ENGINE BEFORE ADMITTING STEAM TO IT.—SIZE OF PULLEYS AND SPEED OF SHAFTING.

Sometimes an engine, especially if it be a small one, is shipped from the factory all ready to be put on to the foundation and belted up for use. In the absence of definite information, the proprietor would naturally ask the engineer that is to have charge of the machine how fast it is to run, and an inspector who is conducting an examination may ask a question, the reply to which will determine whether the engineer understands such matters or not.

Of course any one could turn on the steam after the engine is ready to start, and when it is running at full speed count the revolutions per minute and report the speed, but this may have to be decided on before the engine can be tried in this way.

If the engineer will look the governor over carefully, he will find the number of revolutions that it is calculated to make stamped on it somewhere, or at least this is the practice of the makers of the principal governors now used.

The cut gears which transmit the motion of the governor shaft to the governor spindle and balls are both the same size, therefore they both travel at the same speed. Now it should be remembered that although the engine furnishes the

power to run the governor, still the governor determines the speed at which the engine runs. From this it will be seen that the speed of the governor pulley is the standard, and from this we must make our calculations.

Let us suppose for illustration that the governor pulley is seven inches in diameter, and is to revolve 150 times per minute, and the pulley on the shaft is ten inches in diameter. How fast will the engine run? $150 \times 7 \div 10 = 105$ revolutions per minute.

Sometimes there is no governor pulley on the engine but the required speed is known, and the question is as to what the diameter of the governor pulley should be. Suppose that the engine is to run 90 revolutions per minute, and the pulley on the shaft is 18 inches in diameter, and the governor is stamped 120; $90 \times 18 \div 120 = 13.5$ inches, which is the diameter of the pulley needed for the governor.

But if we knew the speed of the engine and wish to find the speed of the governor the calculation would be $90 \times 18 \div 13.5 = 120$ revolutions per minute. By the above examples, the workings of the following rules are illustrated.

To determine the speed of the driven pulley, multiply the speed of the driver by its diameter, and divide the product by the diameter of the driven. The quotient will be the required speed.

To determine the diameter of a pulley required to run a shaft at a given speed, multiply the speed of the driver by its diameter, and divide the product by the speed of the driven. The quotient will be the required diameter.

If there are two lines of shafting already located with the pulleys on, and the necessary

speed of the driven is known, and the required speed of the driver is desired, then multiply the speed of the driven by its diameter, and divide the product by the diameter of the driver. The quotient will be the speed of the driver. These rules will enable the engineer to calculate the speed of any machine or line of shafting in the mill or factory, excepting that it does not take into account the slipping of belts, and no rule can be formulated that will determine this accurately.

These rules for calculating the speed of shafting are based on the circumference of the pulleys over which the belt runs, although only the diameter is mentioned. This is done for simplicity, as when the diameters are used in both cases, the result is the same as if the circumference were brought into the calculation.

Suppose that we have one line of shafting running 90 revolutions per minute on which is a pulley 36 inches in diameter. From this pulley, power is transmitted to another line of shafting on which there is a 24-inch pulley. The belt connecting the two is 28 feet, three inches or 28.25 feet long. Now if we take off the belt and stretch it out on the floor, and take a pulley that is 36 inches in diameter, and roll it along lengthwise of the belt, when we have traversed the whole length of it, our 36-inch pulley will have revolved three times. If we take another pulley that is 24 inches in diameter and roll it over the belt in the same way, we shall see that it will revolve 4½ times, therefore while the large pulley revolves three times when on the shaft, the smaller one revolves 4½ times or 50 per cent. more. Fifty per cent. of 90 is 45, and 90+45=135, which is the speed of the 24-inch pulley.

Or it may be done as an example in proportion

as follows: The circumference of the 36-inch pulley is 9.42 feet, and that of the 24-inch one 6.28 feet, and the larger one revolves 90 times per minute; therefore 6.28: 9.42: :90: 135. Again, if we work it out according to the first rule given for this purpose, we have $90 \times 36 \div 24 = 135$, which demonstrates that all of the three ways are correct. In this example we can readily see the end from the beginning without any figuring, but others involving much different proportions may be worked out and proven in the same way.

CHAPTER XI.

HOW TO CALCULATE THE MEAN EFFCTIVE PRESSURE.—SEVERAL KINDS OF CRANKS.

In determining the horse power of an engine, three factors are necessary, namely: area, speed and pressure. The first two are easily ascertained, and so is the third, provided you have an indicator to do it with, but if an engineer is requested to tell how he would decide this point, he should reply that it can be done if the initial pressure, point of cut-off and back pressure are known, or at least a good estimate of it may be given. There are two ways of doing this, and one of them is to take the given data, and from it lay out an indicator card, and then determine the mean effective pressure from it, but as the matter of laying out the theoretical expansion line will be spoken of in detail in another part of this book, it will not be explained here. The other way to do it is by calculation as follows:

The initial pressure and point of cut-off being given, divide the length of the stroke in inches by the number of inches traversed by the piston before the steam is cut-off. The quotient will be the ratio of expansion. Find the hyberbolic logarithm of this quotient, and add 1 to it. Divide the sum so obtained by the ratio of expansion, and multiply the quotient by the absolute initial steam pressure. From this product sub-

tract the absolute back pressure, and the remainder will be the mean effective pressure.

For illustration let us take the case of an engine taking steam at 80 pounds pressure, maintaining it up to the point of cut-off, which takes place at one-quarter stroke, and assuming that the valves and piston are tight. If the stroke of the engine is 36 inches, and the cut-off at one-quarter stroke, then 36÷4=9 inches, which is the distance travelled at quarter stroke; 36÷9=4, which is the ratio of expansion. The hyperbolic logarithm of 4 is 1.386, and adding one to it gives us 2.386, and 2.386÷4=.5965. The gage pressure being 80 pounds, the absolute pressure is 80+14.7 =94.7 pounds; .5965×94.7=56.488 pounds average absolute pressure. As we assume in this case that there is no back pressure above the atmosphere, we must subtract the atmospheric pressure, and 56.488−14.7=41.788 pounds mean effective pressure. A proof of the correctness of this rule will be given in the part treating of indicator cards. Some further explanation of the above may be of interest. The term "ratio of expansion" means the number of times that the steam is expanded or increased from its original volume. Thus if the cut-off takes place at one-third stroke, by the time that the piston has reached the end of the stroke the steam occupies a space three times as large as it did when cut off (excluding clearance), and the ratio of expansion is said to be three. If the cut-off takes place at one-half stroke it is two, etc. Hyperbolic logarithms are a series of numbers by which arithmetical calculations are simplified. The logarithm of a number is the exponent of a power to which 10 must be raised to give that number.

The number 10 is used because it is a conve-

nient one, and not from necessity, as other numbers might be taken as a base. To find the hyperbolic logarithm of a number, multiply the common logarithm by 2.302585. Fortunately the books of reference which engineers use contain these in tabular form, so that no one thinks of figuring them out in ordinary practice. In the above example the clearance is not taken into account, but when the rule is applied to any particular case the clearance in that case must be added to the stroke, and also to the space traversed by the piston before the cut-off takes place, and it should be expressed in decimals of an inch. If the engine has a fixed or an adjustable cut-off, the point of cut-off may be determined by taking off a cylinder head, admitting steam to the steam chest, and turning the engine by hand until the steam is shut off by the valve, when the distance may be noted on the guides. This should be done when the cylinder is well warmed up, and the piston rings may be tested by admitting steam to the opposite end, when the engine is on the centre. In the case of a slow-speed, long-stroke throttling engine, the initial pressure may be determined by attaching a steam gage to the steam chest, taking care to put a globe valve in the pipe, so that it may be partially closed in order to prevent fluctuations of the pointer.

The writer does not claim that the mean effective pressure can be accurately determined in any other way than by using a good indicator, but an approximation of it may be obtained as above stated. The elements of uncertainty are the tightness of the valve and piston under actual working conditions, the amount of compression, and also the exhaust opening. We all know what a crank is (although there are several kinds of

them), but if we were asked to tell what it is, some of us might be at a loss for a term to express our meaning, unless we should remember that it is a device for converting the reciprocating motion of the cross head into the rotary motion of the shaft. Some people evidently believe that there is a great loss of power in using the crank, and have made efforts to introduce some other device that will answer the same purpose, but so far without success, and as a consequence the expression that "the man who tries to improve upon the crank is a greater crank than the crank ever was," is sometimes made use of.

The term "valve gear" is used to designate all of the rocker arms, rods, cranks, etc., which form the connection between the eccentric and the steam and exhaust valves. A misunderstanding concerning this is sometimes the cause of ludicrous mistakes, as, for instance, when a certain man was told that the valve gear of the engine needed repairs, he thought that the bevel gears on the lower part of the governor needed attention, and when told that they were all right, said that they were the only gears on the machine.

CHAPTER XII.

RULES FOR DETERMINING THE WEIGHT OF FLY WHEELS.

We have never discovered just why it was that a fly wheel received its name, but the fact remains that a wheel put on the crank shaft to answer the double purpose of helping the engine to run steady, and also to receive a belt for transmitting power, is so called. As the engine begins its stroke, it receives the steam at nearly the full boiler pressure. Soon it is cut off and begins to expand down, until when near the end of it there is but little pressure left, if the load is a light one.

If there were no wheel on the shaft of a single engine, it would run at a very irregular speed, if it revolved at all, but as it is in practice the wheel causes the power stored up at the beginning of the stroke to be given out during the latter part of it, the result being economy in the use of steam and steady speed.

After looking over the several specimens of fly wheels put on engines by their builders, we are caused to wonder if there has been any rule followed determining their size and weight. This is an open question, but nevertheless there are rules for this purpose which will give good results, one of which is as follows: $7{,}000{,}000 \div R^2 \div D = P$, in which R=revolutions per minute, D=diam-

eter of wheel, and P the number of pounds per horse power required in the rim of the wheel.

This rule recognizes the fact that the efficacy of a wheel increases, not as its velocity, but as the square of its velocity and as its diameter. This is shown by the way that the constant number 7,000,000 was obtained, which is as follows: An engine having a fly wheel which gave good results at a known load was taken, and dividing its weight by the horse power developed, and multiplying the quotient so obtained by the square of the number of revolutions per minute and by its diameter.

Let us now explain and illustrate this rule, beginning by reading the formula as follows: Divide 7,000,000 by the square of the number of revolutions per minute, and the quotient by the diameter of wheel. The quotient so obtained will be the number of pounds per horse power required for the rim of the wheel.

For our example let us take the engine spoken of in Chapter 8, in which the diameter of cylinder is 18 inches, the stroke 42 inches, speed 65 revolutions per minute, mean effective pressure 35 pounds, and horse power developed 121.37. It will be necessary to assume the diameter of the wheel, and for convenience sake we will call it 15 feet. Applying the rule our first step is to square the number of revolutions per minute, and $65 \times 65 = 4225$. $7,000,000 \div 4225 = 1656.8$, and dividing this by the diameter of wheel we have $1656.8 \div 15 = 110.45$. The power developed is 121.37, and $110.45 \times 121.37 = 13,405.3165$ pounds. This it must be remembered is for the rim of the wheel.

Let us look for a moment at the foundation on which this rule is based. Let us suppose that this fly-wheel had been made for this engine, and

is found to give good results. Taking this as a basis, we divide the weight by the power developed, and multiply by the square of the number of revolutions, and by the diameter, 13,405.3165÷121.37×4225×15=6,999,768.75. This is not quite as much as the constant number given, but the difference is due to the fact that the fractions are not carried out indefinitely.

There is another rule which may be preferred by some, and it is used by many of our designers, as follows: Divide 12,000,000 by the square of the diameter of the wheel in feet, and this quotient by the square of the number of revolutions per minute, and multiply this quotient by the area of the cylinder in square inches and by the stroke in feet.

Applying this rule as before, we have 12,000,000 ÷225÷4225×254.46×3.5=11,221 pounds. This it should be remembered is to be the weight of the rim of the wheel, and is said to be sufficient to give good results for ordinary cases, but where the work is variable, as in rolling mills and some saw mills, the former is preferable, or may be exceeded, but it is well to bear in mind the fact that while in most cases where iron is used, to increase the amount of material used is to increase the strength of the machine, yet this does not apply to the rim of a fly wheel, owing to the fact that when the weight is increased the tendency to burst is also increased.

It may be well to speak of a rule for determining the entire weight of wheel, and the following is one said to be used by one of our most prominent engine-building firms: $785{,}400 \frac{D^2 \times S}{D^2 \times R^2} = W.$ d=diameter of cylinder in inches, S=stroke in

inches, D = diameter of wheel in feet, R = revolutions per minute.

This formula signifies that the constant whole number 785,400 is to be multiplied by a fraction whose numerator is found by multiplying the diameter of the cylinder in inches by itself, and the product by the stroke in inches.

The denominator is determined by multiplying the diameter of the wheel in feet by itself, and also by the square of the number of revolutions per minute. The final product will be the weight of the entire wheel.

Applying this rule or formula to the case in hand we find the square of the diameter of the cylinder in inches to be 324. The stroke is 42 inches, and $324 \times 42 = 13,608$, which is the numerator of the fraction. The diameter of the wheel is 15 feet, and the revolutions per minute 65, therefore $15 \times 15 \times 65 \times 65 = 950,625$, which is the denominator, and the fraction then reads

$$\frac{13,608}{950,625}$$

and by dividing each part by nine, or in other words by cancellation, it may be reduced to

$$\frac{1512}{105,625}$$

In order to multiply our constant whole number by this fraction, we must multiply it by the numerator and divide the product so obtained by the denominator: $785,400 \times 1512 \div 105,625 = 11242.8$ pounds for the entire wheel.

We give these rules just as we find them, trusting that the examples have been worked out in such a way as to be plain and easily understood by the engineer of limited education, in order

that he may make comparisons for himself. The first one is from a well known authority, while the second and third are from another source usually accepted as reliable.

The results obtained differ widely, and it is proper that they should, for no one rule will apply to every case, for if we have a long-stroke, slow-speed engine, running a rolling mill or a sawmill, or any other kind of a mill where the load varies greatly, it will need a heavier wheel than a high speed engine running a dynamo or any other machinery that operates in such a way as to make the load practically constant will; therefore, when a wheel is designed, the kind of work that it is intended for must be taken into consideration.

If the rim of a sound wheel is never run at a greater speed than 5000 feet per minute it will be within safe limits, but in designing a plant it would be well to make it less than this if possible.

All fly wheels should be carefully balanced before being put into service, as it will tend to cause the engine to run steadier, the bearings to run cooler, and the wheel will be safer. In bolting together the sections of a wheel, good judgment should be used, in order to avoid unnecessary stress in screwing up the nuts.

CHAPTER XIII.

BEFORE AND AFTER ADDING A CONDENSER.—IS THE MEAN EFFECTIVE PRESSURE EFFECTED BY IT?

Reference has been made in a preceding chapter to the practice of adding a condenser to a non-condensing engine, and as the candidate for a license will probably be called upon to answer some questions concerning this appendage to an engine, he should give the matter some attention.

In a certain work on steam engineering, which lies before me, I find the statement that in ordinary practice if we add a condenser we shall raise the mean effective pressure by 12 pounds. This authority does not tell us what becomes of this increase of 12 pounds, or how we can use it up.

A careful consideration will show any one that to run a certain machine will require a certain mean effective pressure in the cylinder, and whether we have a condenser in use or not does not affect this pressure in the least. If we start up another machine the mean effective pressure will be increased at once, and it will be just the same whether we use a condenser or not. This refers to ordinary practice.

If, however, we had an engine using steam whole stroke, and maintaining boiler pressure up to the end of the stroke, running non-condensing and without a governor, then by adding a condenser we could increase the mean effective presssure by putting on a condenser, but this is not

ordinary practice, and in fact is seldom, if ever, found in every-day use.

Assuming this to be the case, the percentage of gain in pressure may be calculated as follows: Multiply the reduction in back pressure by 100, and divide by the mean effective pressure. Suppose that our boiler pressure is 80 pounds, and our mean effective pressure 80 pounds also. We reduce the back pressure by 12 pounds and $12 \times 100 = 1200$, and dividing 1200 by 80 gives us 15, which is the percentage of gain in mean effective pressure. The above conditions are the only ones under which a condenser adds to the mean effective pressure.

In ordinary practice, if our initial pressure is 80 pounds and our cut-off at one-third stroke, if we add a condenser and thereby reduce the absolute back pressure by 12 pounds, our governor immediately reduces the forward pressure by 12 pounds, and as the mean effective pressure is always the difference between the forward and the back pressure, and they are both reduced alike, the mean effective pressure is just the same as it was before. This is a theory that has been proven to be correct in practice, many times over.

If it is desired to know how much the addition of a condenser will allow us to reduce the boiler pressure, and maintain the original point of cut-off, the following is a simple formula for expressing it:

$$\frac{P \times S}{p' \times (H.\ P. + 1)} = P'$$

in which P = mean effective pressure, S stroke in feet, p' = point of cut-off, plus clearance expressed in feet, H. P. = hyperbolic logarithm of ratio of expansion, P' = boiler pressure.

It signifies that the mean effective pressure multiplied by the stroke in feet will give us what we may term our first product. The point of cut-off, or in other words the distance travelled by the piston before the cut-off valve closes, in feet, plus the clearance, multiplied by the hyperbolic logarithm of the ratio of expansion plus 1, will give us our second product. Dividing the first product by the second gives us the desired boiler pressure.

Now if we assume that the initial and boiler pressure are both 80 pounds, the cut-off at one-quarter stroke, and not taking the clearance into account, which will affect the result but a very little, the mean effective pressure will be 41,788 pounds. The stroke is 3.5 feet and the cut-off is at one-quarter stroke. 3.5÷4=.875. The ratio of expansion is 4, the hyperbolic logarithm of which is 1.386 adding 1 to it equals 2.386. The problem is then as follows:

$$\frac{41.788 \times 3.5}{.875 \times 2.386} = 70 \text{ pounds.}$$

It is assumed that the condenser will reduce the absolute back pressure by 12 pounds.

In case that the question is asked, as to how much the consumption of steam will be reduced by adding a condenser, maintaining our initial pressure, the answer is expressed in the following formula, assuming the ratio of expansion to be six. L÷R=1, in which L equals length of stroke in feet, R equals ratio of expansion (assumed), 1 equals point of cut-off in feet. Applying this to the case in hand, we find that 3.5÷6=.583 feet, or .583 of 12 inches equals 6.996 inches. By this we see that whereas the steam followed the piston for 10.5 inches of the stroke before the condenser

was made use of, it is now cut off at 6.996 inches, and therefore the volume admitted is less in direct proportion.

If we were to calculate the amount saved figuring from these two volumes, we should have a simple problem in proportion, as the whole stroke would be taken as 100. 10.5: 6.996: :100 :66.6. This is solved by multiplying 6.996 by 100 and dividing the product by 10.5. In other words, the consumption of steam would be reduced from 100 to 66.6 per cent., a saving of 33.4 per cent. on the face of it, as we sometimes say.

It is possible that this would not be realized in practice, for while the volume admitted, as shown by the indicator, is as above stated, still, as the terminal pressure is less, the condensation would be more, which of course would effect the saving in coal.

In regard to the formula for determining the point of cut-off we would say, that as we have but one factor to start with, namely the length of the stroke in feet, and from this we must ascertain the other two, it is necessary to assume one of them and try it. If it does not prove to be correct we must try another. In this case we assumed the ratio of expansion to be six, and the way to prove whether it is correct or not is as follows: Taking the data which we now have, we must determine the mean effective pressure from it, by the rule given in Chapter 11 of this book. The cut-off is at one-sixth stroke, therefore the ratio of expansion of 6. The logarithm of $6+1=2.7918$ and dividing by six we have .4653 as a quotient. The initial pressure is 80 pounds by the gage, to which we must add the atmospheric pressure; $80+14.7=94.7$, and $.4653\times94.7=44.06$ pounds mean pressure. If the condenser re-

moves 12 pounds of the back pressure due to the atmosphere, we shall 2.7 pounds left, and 44.06 − 2.7 = 41.36 mean effective pressure. As it originally was 41.788 pounds, the difference is but a fraction of a pound, and is practically identical. If it was not we should have to assume another ratio of expansion and try again. The author would state, in parenthesis, that he has indicator cards in his possession which prove these conclusions to be correct in a remarkable degree.

CHAPTER XIV.

SAFE WORKING PRESSURE OF A STEAM BOILER.

Having devoted a large portion of the preceding chapters to answering questions concerning the engine, we will now turn our attention to the boiler, but in doing this what a transformation we have made. From the contemplation of the bright, symmetrical engine, we turn to the black, sooty and dirty boiler, but the man who aspires to be a licensed engineer and to hold a first-class certificate to that effect, will find that there are many questions to be answered concerning the "source of power," and that a knowledge of its construction, the strength of the materials of which it is constructed, the comparative strength of the seams, of the manner in which it is braced, and other points concerning it will be necessary. The first thing that we will take up for consideration in detail will be the way to calculate the safe working pressure of any boiler.

For illustration suppose that a boiler is made of iron three-eighths of an inch thick, the longitudinal seams are double riveted, and the diameter of the shell is 60 inches. To this case the following formula for determining the safe working pressure of a double riveted boiler will apply.

$$\frac{T \times .70 \times 10{,}000}{R} =$$

safe working pressure, in which T = thickness of iron in decimals of an inch, and R = radius, or

one-half of the diameter. Then .375×.70×10,000 ÷30=87.5 pounds safe working pressure. Our first step in solving this problem is to reduce ⅜ to decimals of an inch by setting down 100 and dividing it by the denominator of the fraction, which in this case is eight, and adding ciphers to our dividend (100) until we can divide by eight without a remainder. We find that by adding one cipher our quotient is .125, which is one-eighth of an inch expressed in decimals, and by multiplying by three we have .375.

This may also be done by adding ciphers to the numerator and dividing by the denominator. By adding three ciphers, we can divide without a remainder and by pointing off as many decimals in our answer as we have added ciphers we get the same result, 3,000÷8=.375 as before.

We do this because the strength of boiler plate is estimated at a certain amount per square inch of sectional area, or in other words, a bar one inch square will require a certain amount of stress to cause it to break in two. As our plate is but three-eighths of an inch thick, it is proper that we should take that as a basis for our calculation. We multiply by .70 because the riveted seam is never as strong as the solid plate, and where there are two rows of rivets it is estimated to possess .70 of the strength of the solid plate. We next multiply by 10,000 because good boiler iron has a tensile strength of 50,000 pounds to the square inch of sectional area, and taking one-fifth of this for a safe load gives us the number 10,000.

If the tensile strength is 60,000 pounds and the factor of safety 5 then we should have 60,000÷5= 12,000 instead of 10,000 and if it is 40,000, then 40,000÷5=8,000 pounds. The number that we

divide the tensile strength by is called the factor of safety.

We divide by one-half of the diameter, because in this case we have calculated the strength of but one side of the boiler, taking one inch of its length and therefore must not calculate on the whole diameter. This rule is probably as good a one as can be found, as it takes into consideration all of the elements in the case except one, and that is it does not recognize the fact that all double-riveted seams do not possess the same strength. It is given to us in this form because Fairburn found by experimenting with double riveted seams that 30 per cent. of the strength of the solid plate was lost in punching, but this does not go to show that all seams are alike, simply because they have two rows of rivets in them.

Having given a rule which the writer considers to be a good one, he wishes to speak of some others, and express an opinion concerning them. Here is one that is published by a firm whose name is calculated to inspire confidence. Multiply twice the thickness of the iron by the tensile strength of sheet, and divide the product by diameter of shell in inches. Divide this product by six. For double riveted seams add 20 per cent. This rule apparently assumes that it is safe to carry but one-sixth of the bursting pressure in every-day practice, but when we examine a little closer, we find that it is not as safe as it looks to be. It will be noted that the whole strength of the sheet is taken, no deduction being made for the loss of strength due to the single riveted joint. With this rule the factor of safety is but 3.5, which we consider altogether too small for the average boiler.

For the purpose of comparison, let us assume

the elements of the preceding case and try it. Thickness of iron, .375; tensile, 50,000 pounds; diameter, 60 inches. Applying this rule, we have $.375 \times 2 \times 50{,}000 \div 60 \div 6 = 104.16$ pounds for a boiler with single riveted seams. Adding 20 per cent. for double-riveted seams, we have $104.16 + 20.83 = 124.99$ pounds for safe load. Would not the average intelligent engineer doubt the expediency of carrying 125 pounds pressure on such a boiler? Another authority gives us the following formula:

$$\frac{S\,B\,t\,2}{D\,C} = P$$

in which

S = tensile strength of plate.
B = strength of joint.
t = thickness of plate.
D = diameter of shell.
C = coefficient of safety.

The "coefficient of safety" means the "factor of safety" which we assume to be 5.

Now this formula, although it looks rather complicated, is really not so when explained. If we assume the elements of the preceding case it would stand as follows:

$$\frac{50.000 \times .70 \times .375 \times 2}{60 \times 5.} = 87\text{-}5 \text{ pounds.}$$

It will thus be seen that this rule is identical with the first one quoted, although it appears in quite a different way.

The following rule is also given us for determing the safe working pressure. Divide the thickness of the plate in inches by the diameter of the boiler in inches, and multiply the quotient by 7600 for an iron boiler with single riveted seams, and 9000 for double riveted seams. Applying

this to the case in hand we have $.375 \div 60 \times 9000 = 56.25$. Probably this is a "safe" pressure to carry on such a boiler, judging by the pressures that are safely carried on such boilers in everyday practice. Here are four rules given us for determining the pressure that it is safe to carry on a boiler of given dimensions, which gives us three widely different results, and which of these are we to adopt? The first and third rules, giving 87.5 pounds for an answer, are undoubtedly the proper ones to adopt. The second one shows the effect of competition as to who can allow the highest pressure to be carried without disaster, while the fourth one is many years behind the times, and is not consistent with good practice.

We cannot tell how high a pressure it is safe to carry until we know the actual strength of the seams based on the size and pitch of rivets, etc. In Chapter 15, due attention will be given to this important factor, as in the preceding cases we have only assumed that the seam possessed 70 per cent. of the strength of the solid plate.

In a certain book, claiming to give correct rules and instructions to engineers and firemen, I find the following: "Rule to find aggregate strain caused by the pressure of steam on the shells of boilers. Multiply the circumference in inches by the length in inches; multiply this answer by the pressure in pounds."

This is incorrect, for while such a rule will give the total pressure on the shell of a boiler, it is not a proper way to determine the strain (stress) on the shell, as that is determined by multiplying the diameter in inches by the length in inches.

CHAPTER XV.

STRENGTH OF THE SEAMS IN A BOILER.

In considering the strength of any riveted seam, it is very plain that it cannot be taken at anything more than the weakest part of it. This may be in the rivets or it may be in the plate, or rather what is left of it after it is punched or drilled. No joint can possess more strength than that due to the material left between the first row of holes, for it will make no difference whether there are one or two additional rows of rivets. The double welt butt joint owes its efficiency to this fact largely, for in this case the rivets in the first row may be set wide apart, thus increasing the net section of plate at this point.

Suppose that we take up the illustration used in Chapter 14, in which the plate was 3-8 or .375 thick, the tensile strength of it 50,000 pounds, and assume the pitch of the rivets to be 3 1-16 inches (3.0625 inches) their diameter, when in position, 15-16 or .9375 inch. Now the pitch of the rivets mean the distance from the centre of one rivet hole to the centre of the next one, therefore when we wish to calculate the strength of this section of the plate, before the holes are punched, we must multiply the width of it by its thickness, which will give us the sectional area, and by multiplying the product so obtained by the tensile strength of iron, we shall have the strength of the section of solid plate.

After the holes have been punched, as one-half

of each hole is included in the section designated by the pitch of rivets, we must deduct the diameter of one hole from the section, when we wish to calculate the strength of what is called the net section of plate. Illustrating the above ideas we have 3.0625×.375×50,000=57,421 pounds, which is the strength of the solid plate. Deducting the diameter of one rivet hole, we have 3.0625—.9375 ×.375×50,000=39,843 pounds, strength of net section of plate. As we have now determined the strength of plate, at seam, let us calculate the strength of the rivets in the joints. In the section that we have taken there are two rivets included, for although in taking the pitch of the rivets, we include but one-half of two rivets in the first row, still, as there is another rivet directly between these two located in the second row, we must calculate on two rivets. The area of one is .69 and 69×2=1.38. If we take the strength of the iron in the rivets as being the same as that in the plate we must multiply 1.38 by 50,000 and our product is 69,000 pounds, which is the strength of the rivets.

Again we find a difference of opinion among experts, for while some tell us that the iron in the rivets should be taken as the same as that in the plate, others claim that it is less, putting it about .8 of it. If we should adopt this idea, we must multiply 69,000 by .8, giving us a product of 55,200 pounds as the strength of rivets. The net section of plate is therefore the weakest, and by dividing 39,843 by 57,421 our quotient is found to be .69, or in other words, our double-riveted seam possesses 69 per cent. of the strength of the solid plate. In our original calculation we assumed it to be 70 per cent., which is near enough for practical purposes, under the

conditions named, but if the rivets were larger or smaller, or had a different pitch, the results would be different as a matter of course. By this we see that all the details of the joint must be given before we can tell the strength of it.

Another rule for determining strength of net section of plate is expressed by the following formula:

$$\frac{P-D}{P}=\text{Plate},$$

in which P equals pitch of rivets and D equals diameter of holes. Applying this to the case in hand, we have

$$\frac{3.0625-.9375}{3.0625}=69 \text{ per cent. of solid plate.}$$

Another rule for determining strength of rivets is expressed by the following formula:

$$\frac{A\times R}{P\times T}=\text{Rivets},$$

in which A equals area of rivets, R equals number of rows, P equals pitch of rivets and T thickness of plate. Applying this to the joint that we have been considering, we have:

$$\frac{.69\times 2}{3.0625\times .375}$$

equals 1.2 times the strength of solid plate.

We have already shown that the strength of the section of plate is 57,421 pounds and 57,421×1.2= 68,905 or very nearly 69,000 pounds as before.

This shows that the net section of plate is the weakest, as did our former calculation. These rules do not apply to the double welt butt joint, as now made for boilers of large diameter,

calculated to carry high steam pressures, for the first row of rivets are spaced much farther apart than is ordinarily done, or in other words given a greater pitch.

The effect of this is to greatly increase the strength of the weakest part of the joints, namely the section of plate between the rivets, in the first row. Under any circumstances liable to be found in practice, the rivets in this joint are much stronger than the plate is, but if it is desired to calculate the strength of rivets it may be done by taking a section extending from the centre of rivet in the first row to the centre of the next one in the same row, and estimate it as a single-riveted joint. To this must be added the strength of the rivets in the second and third rows, for a section of the same width, and add the two together. In order to familiarize himself with these calculations, the candidate for a license should practice on different problems along this line, supplying data himself.

CHAPTER XVI.

BRACING FLAT SURFACES IN A BOILER.

In the ordinary tubular boiler, the longitudinal seam is the one by which the strength of the whole structure is usually calculated, for the tubes strengthen the heads greatly, and from the nature of the case the curvilinear seams are subjected to much less stress than the others are. There is, however, a space on each head above the tubes, which must be held by braces, aud also a space below them in those boilers which have a manhole in lower part of front head, as all of them should have. It is a very good plan to have these braces extend from one head to the other, as they strengthen the shell in that case, without adding any unnecessary stress to the iron in the braces. Another reason is that they then are in a position to hold to the best advantage, as a brace standing at an angle greater or less than 90 degrees cannot hold as much stress on the heads as one that stands at right angles to the heads. Where they are put on from head to shell, their efficiency decreases as their length is made less, which fact is not always taken into consideration, judging by the way some boilers are constructed.

In designing this part of a tubular boiler, it is necessary to take into consideration the surface to be braced, the pressure that it will be called upon to withstand, and the size of the braces. We are told that braces in a steam boiler should not be made to withstand a pull exceeding 6,000

pounds per square inch of sectional area. If braces were made of iron one inch square it would be much less trouble to estimate their capacity, but as they are usually made of round iron, it is necessary to reduce them to square inches. A piece of round iron 1⅛ inches in diameter contains a square inch of iron, practically, but if a brace is made of round iron 1.5 inches in diameter we must ascertain its area by squaring its diameter and multiplying by .7854, thus: $1.5 \times 1.5 \times .7854 = 1.767$ square inches. Now if each square inch of section may be depended upon to bear a stress of 6,000 pounds, in order to ascertain how much it will do to put on a brace 1.5 inches in diameter, we must multiply 1.767 by 6,000, and our product is 10,602 pounds.

Suppose that we find that we have a flat surface on one of our boiler heads, containing 700 square inches of surface, and we wish to carry 90 pounds of steam pressure. By multiplying 700 by 90 we find that we have a load of 63,000 pounds to provide for, and as each brace will hold 10,602 pounds, by dividing 63,000 by 10,602 our quotient is found to be 5.94, therefore we shall need to put in six braces, in order to make the head safe.

Braces should be very carefully inspected before being put into service, in order to guard against defects in the iron, especially where they are welded, and we note with pleasure that braces are now in the market that have no welds in them, as we believe that this is a step in the right direction.

We wish to call attention to the fact that in calculating the stress on a brace extending from head to head, it is proper to take the surface on one head only, and not on both of them. This does not seem to be generally understood by en-

gineers, although much has been said on the subject. The writer recalls an instance where he made this statement before a society of engineers, and one of them remarked in an exceedingly sarcastic manner that in his boiler there was a pressure on both of the heads. Well, what of it? We never doubted the truth of the statement, but as that fact has no bearing on the idea that in calculating the load on the brace, we must take the surface on one head only, we made no reply.

If the brace extends from head to shell, there is just as much stress on it as if it extended from head to head. A brace should never be put from head to shell on the lower part of a tubular boiler, for where it is riveted to the shell it makes a place for scale to collect, and the flat part of a brace so put in prevents the water from coming in contact with the iron of the shell, and that too in about the hottest part of the boiler.

It is customary to rivet pieces of T iron to the flat head to be braced and then attach the braces to them. This, of course, greatly strengthens the heads, and makes a good anchorage for one end of the braces.

It may not be out of place here to remark that where the rivet holes are drilled the sheets are stronger than where they are punched, because punching disturbs the fibres of the iron, but when drilled the edges of the holes are more sharply defined, therefore the rivets will shear more easily.

However, the difference is in favor of the drilled holes. In calculating the area of a boiler head to be braced it is not necessary to take into consideration the whole surface exposed to pressure, for the flanges impart stiffness to it, to a limited

extent, according to the thickness of the head. If it is 9-16 or 5/8 inch thick it will do to leave a space of two inches from the shell, out of the calculation. This will receive further consideration in another chapter.

CHAPTER XVII.

THE HORSE POWER OF A BOILER.—PRIMING AND FOAMING.

This book would not be complete unless it contained instructions as to the proper way to calculate the horse power of a boiler. We occasionally see it stated in print that in the case of a tubular boiler 15 square feet of heating surface constitutes a horse power. This may be correct and it may not be, for it will depend much on other conditions. In a preceding chapter the proper way to calculate the horse power of an engine was illustrated and explained. From the data there given it will be seen that while the size of the cylinder is an important factor in solving the problem, still that of itself does not determine the horse power of the engine, but other things must be taken into consideration.

The square inches on the inside surface of the cylinder does not tell us the power of the machine, neither does the square feet on the outside surface of a boiler tell us the power of it, for in either case it will depend on how much heat is applied, etc. The only proper way to determine the horse power of a boiler is by the amount of water that it will evaporate per hour under given conditions. It will evidently make a difference whether this water is to be evaporated into steam of 10 pounds pressure or of 150 pounds pressure, for in one case it will require much more heat than in the other.

Again, it will make a difference whether the water to be converted into steam is hot or cold when it is delivered into the boiler. If it is put in at a temperature of 40° it will need more heat to convert it into steam than it will if it enters at 212°. From this it will be seen that some standard must be adopted in order to bring all of the different conditions under which boilers are run, to a common basis for the purpose of comparison. The standard usually adopted is the evaporation of 30 pounds of water per hour from a temperature of 100° into steam at 70 pounds pressure. Now a good tubular boiler, well set and properly run, will develop a horse power for each 15 square feet of heating surface without doubt, and some of them are developing nearly two-horse power for each 15 square feet of heating surface, while others require 20 or more. Some firms who manufacture boilers allow one square foot of grate surface for each 36 feet of heating surface, and this works very well in practice.

The candidate for a license should bear in mind that no inflexible rule can be given for determining the heating surface of a boiler, as, for instance, some writers say take one-half of the surface of the shell of the boiler and two-thirds of the heads. This is evidently a mistake, although perhaps not a very serious one in its consequences, for if the brick setting does not come in contact with the shell until it is above the centre line of boiler, then more than one-half of the shell is available as heating surface. Many boilers are set in this way. To this should be added the internal surface of all of the tubes or flues. Again the area of one head up to the water line should be determined, and the combined area of the holes cut for the tubes subtracted from it.

This will give the heating surface in the heads when multiplied by two.

The water space of the boiler is that part of it which is occupied by the water, and is found by ascertaining the area of one head up to the water line, as before referred to, and subtracting the combined area of the holes cut for the tubes. Multiply this by the length of the boiler in inches and divide by 1728. This will give the number of cubic feet of water in it. To change to gallons multiply by 7.5. To find weight multiply the number of gallons by 8.33. The steam space of a tubular boiler is found by ascertaining the area of one of the heads above the water line, and multiplying it by the length of the boiler in inches. Divide by 1728. If there is a dome the cubical contents must be added to the above. For rules to determine areas of segments of circles, see Chapter 48.

When an engineer says that his boiler "foams," it is generally understood that he means that the water froths up into the steam space, so that it is almost impossible to tell where the true water level is. It is a dangerous fault, and is usually caused by foul water or too small a steam space. Sometimes changing from salt to fresh water, or vice versa, will cause a boiler to foam. When we say that a boiler "primes," we usually mean that more or less water is continually carried off with the steam, causing it to be what is called "wet steam." When the report of an evaporative test is given, it is unreliable unless the condition of the steam is known, for water carried off in this way is not evaporated, but when it is so counted, causes the boiler to be credited with greater efficiency than it is entitled to.

If an upright boiler primes it may sometimes be

stopped by removing some of the tubes, thus increasing the steam space. If it is a tubular boiler, and the priming is excessive it may be stopped by putting in a dry pipe. Priming is a source of loss of fuel, and may result in injury to the engine. Nearly all boilers prime more or less, and it may go on undetected for years, unless the steam is tested, as the glass gages give no intimation of it.

Generally boilers furnishing steam for automatic engines prime less than if the engine is a throttling one, taking steam nearly whole stroke. A separator is a device forming a part of the steam pipe, and is applied for the purpose of catching the water in the steam, and preventing it from going to the engine. Water so entrapped may be returned to the boiler at a high temperature.

CHAPTER XVIII.

SAFETY VALVES.

There are several rules relating to safety valves which an engineer who hopes to receive a first-class license should be familiar with, the first one that we shall call attention to being the one, or rather several, for determining the proper size of valve for any boiler. As we believe that the rule adopted by the United States Government is a very good one, we shall speak of that first, and it says that we must allow one square inch of safety valve area for each two square feet of grate surface, in the case of a common lever valve, and allow one square inch of valve area to each three square feet of grate surface, if the pop valve is adopted. This refers to boilers using natural draught, and for such it will be ample.

We believe that the grate surface is the proper base for the calculation, rather than the heating surface, for if the grate surface is large in proportion to the heating surface, then if the size of valve was calculated from the latter, it might not be sufficient for the purpose intended, and become a danger valve instead of a safety valve. The English rule is to allow one-half square inch of valve area for each square foot of grate surface, which is identical with the above for natural draught and lever valve. Another one said to be Prof. Rankine's is to allow .006 of a square inch of valve area for each pound of water evaported per hour.

It is rather a difficult manner to design a valve

by this rule, for who can tell how much water the boiler will evaporate in advance?

We could give what might be called "a shrewd guess," but that would not fill the bill.

Another one, said to be the French rule, is as follows: Multiply the grate surface in square feet by the number 22.5. This gives number one. Add the number 8.62 to the steam pressure allowed. This gives number two. Divide one by two and the quotient will be the area of the valve in square inches. This rule embodies a correct principle, for it takes into account the steam pressure to be carried, which it may be well to do, for steam of a high pressure will escape into the atmosphere more rapidly than steam of a low pressure will, the openings being the same size, therefore we shall need a larger valve for a boiler that is to carry a low pressure of steam.

It may be well to note, however, that the rule is defective in that the first number given should be much larger than it is to give correct results.

For illustration, suppose that we have a 54-inch boiler with a grate surface of 25 square feet, to carry 80 pounds steam pressure. Applying the rule, we have $25 \times 22.5 = 562.5$; $80 + 8.62 = 88.62$; $562.5 \div 88.62 = 6.34$ square inches valve area, which is just about one-half what it should be.

A rule adopted by the German government takes into account the heating surface of boiler, and pressure to be carried, but as it is not complete without a table it will not be inserted here. If the candidate gives the first rule mentioned probably it will be all that will be required of him.

We think that it is not generally understood that a valve having a bevelled seat will not give as large an opening as one having a level one will, and at first thought it will appear as if it would be

the same in both cases, for although the valve and seat may be on an angle of say 45°, does not the valve lift just the same, thus bringing all parts of it at the same distance from the seat as if there was no bevel to be taken into account?

If any one who is interested will draw a sketch of a valve and seat on an angle and represent the valve as lifted one-quarter inch from its seat, they can easily see that the distances measured at right angles to the seat, between the valve and the seat, is less than one-quarter inch, and as this is what determines the area of an opening, it must of necessity be less than a flat valve and seat would give. To determine what the area of opening will be the following rule is given, provided the angle is at 45°: Multiply the diameter of the valve by the lift, and this product by 2.22. Multiply the square of the lift by 1.11. Add the two products and their sum will be the required area of the opening of valve in square inches.

The rule for determining the area of opening of any valve, with a bevelled seat, at whatever angle it may stand, provided, of course, that the lift of the valve is less than the depth of the seat, is as follows: Multiply the diameter of the valve by the lift, and this product by the sine of the angle of inclination and this product by 3.1416. This gives what may be termed the first product. Multiply the square of the lift by the square of the sine of the angle of inclination, and multiply the product so obtained by the co-sine of the angle of inclination, and this product by 3.1416. This gives what may be termed the second product. Add the first and second products together and their sum will be the area of the opening in square inches. If both valve and seat were flat, then all that we should have to do would be

to multiply the circumference of the valve by its lift and the product would be the area of the opening in square inches. As these rules may not be fully understood by some, they will be illustrated and fully explained in Chapter 19.

CHAPTER XIX.

MORE ABOUT SAFETY VALVES.—HOT AND COLD WATER PUMPS.

For the purpose of illustrating the rules given in Chapter 18 for calculating the area of safety valve openings we shall give one example, in order that the results may be compared. The author finds that in many works giving these rules, different examples are given for each rule, but this is not at all necessary, and, in fact, is a detriment, for if two rules are given for solving the same problem the results should at least very nearly agree. Let us take a valve four inches in diameter with a seat bevelled to an angle of 45°, and assume that the lift is one-fourth of an inch.

Applying the first rule we find that $4\times.25\times2.22=2.22$ which is our first product: $.25\times.25\times1.11=.069375$, which is our second product. Adding these two together we find that $2.22+.069375=2.289$ square inches. It may be expressed in an abbreviated form as follows: $D\times L\times2.22=A$. $L^2\times1.11=B$. $A+B=R$, in which D=diameter of valve, L=lift of valve, A=first product, B=second product and R=area of opening. Applying the second rule we find that $4\times.25\times.707\times3.1416=2.2211$, which is the first product; $707\times707\times.25\times.25\times.707\times3.1416=.0693$ which is our second product. When this product is written out in full it requires 17 decimal places to express it, but the result is not materially changed by omitting 13 of them. $2.2211+.0693=2.2904$ square

inches area of opening. It may be expressed as follows: $D \times L \times S \times 3.1416 = F$. $L^2 \times S^2 \times C \times 3.1416 = P$. $F + P = A$, in which D=equals diameter of valve, L=lift of valve, S=sine of angle of inclination, C=co-sine of angle of inclination, F=first product, P=second product, A=area in square inches. If the valve and seat were flat, then we could make use of the third rule as follows: $12.5664 \times 25 = 3.1416$ square inches.

This rule may be expressed as follows: $C \times L = A$, in which C=circumference, L=lift and A=area of opening in square inches. We wish to call attention to the fact that the valve with a flat seat gives 37 per cent. more opening than the one with a bevelled seat, at a 45-degree angle, the lift being the same in both cases, and as the angle decreases (from the perpendicular) the opening grows less.

The candidate should be familiar with rules for determining the pressure at which a valve will blow off at under stated conditions; for telling the length of lever for a certain valve, and also the weight required on a lever of given length to allow the valve to open at a stated pressure, and also should be ready to assume data for examples illustrating all of these rules.

He should also look up matters in connection with boiler feeders, and one of the most difficult questions to give a proper reply to, is how to determine the proper size of pump for a battery of boilers. Not that there is any lack of rules for this purpose, the most prominent one being to provide a pump capable of delivering one cubic foot or 7.5 gallons of water per hour for each 15 square feet of heating surface.

This is usually sufficient, and oftentimes largely in excess of what will be used at any time. How-

ever, it is well to have a pump large enough to do its work easily, for pumps should not be expected to run at the speeds given them in the maker's catalogue, as they are excessive. A good direct-acting pump will run at a very slow speed, and it is well to run it at from 25 to 50 strokes per minute, instead of from 100 to 300 as listed in some catalogues. There is no way to tell in advance just how much water will be needed to operate a plant, but when the amount of coal consumed is known, then under average conditions the number of pounds of coal burned will be the number of gallons required. This of course, would not apply to plants that are used for heating purposes during the winter, and returning the water of condensation by means of some trap, or other automatic arrangement, and for running an engine also, as the amount of water needed in the winter season is practically no more than what is needed in the summer time, although the amount of coal burned may be much greater.

It is a very convenient way to refer to a pump manufacturer's catalogue when it is desired to know the capacity of a pump, but if it is desired it may be calculated by multiplying the diameter of water plunger by itself, and the product by .7854. By multiplying the product so obtained by the length of the stroke and dividing the product by 231 the number of gallons per stroke may be known. The formula is as follows:

$$\frac{A \times L}{231} = G$$

in which A=area of piston, L=length of stroke in inches, and G=gallons contained. If a part of this space is occupied by a piston rod it must be dedncted from the above. If any form of injector

is to be used as a boiler feeder the capacity of it must be determined by experiment, or by consulting the manufacturer's catalogue. It may be well to remember that the capacity of an injector is much greater when the water is delivered to it under pressure than it is when it must lift its supply from a well or tank.

A pump will take water much hotter than an injector (if the supply is above the pump,) because the steam supplying the injector must be condensed and the water must be cold enough to do this. Hot water cannot be lifted as cold water can, because when the pump removes a part of the atmospheric pressure the partial vacuum is at once filled with vapor instead of a solid body of water. The only difference between a hot water and a cold water pump is that if it is to be used for cold water only it is customary to make the valves of soft rubber, which answers every purpose; but if hot water is to be used, then the valves should be made of metal in order to insure durability. Sometimes hard rubber valves are used, but metal ones are better.

It should be remembered that a larger pump will be required to pump hot water for a given plant, than for the same one if the water is to be cold when pumped. Theoretically, a pump that is perfect in every way should just raise a column of water 33.947 feet high at sea level, or rather it should create a perfect vacuum, and then the pressure of the atmosphere on the surface of the water outside of the pipe should force it up, but perfection is rarely obtained in worldly matters, and pumps are no exception to the general rule, therefore they should never be expected to elevate water more than 25 feet, and they will work better if it is even less than this.

CHAPTER XX.

HEATING THE FEED WATER.—PROPER SIZE OF CHIMNEY FOR A STEAM PLANT.

Under ordinary conditions water cannot be heated above 212° by exhaust steam without back pressure on the engine (above the atmospheric pressure). A good form of heater, properly proportioned for the work that it has to do, will heat the water nearly as hot as this in every-day practice, and in some cases water is delivered at the boiling point almost constantly. This saves repairs on the boilers, by avoiding excessive contraction caused by pumping cold water into a hot boiler. This alone will be of enough value to repay the cost of a heater several times over, and we cannot give a rule for calculating just what this saving will be, but the per cent. saved by utilizing a portion of waste heat in the exhaust steam may be determined by subtracting the temperature of the water as it enters the heater from its temperature as it leaves the heater, and dividing the remainder by the total heat of the steam in the boiler.

For an example, suppose that we have water entering at 40° F., and leaving at 212°, and are using steam at 70 pounds pressure; 212−40=172 Steam at 70 pounds pressure contains a total heat of 1210°; 172÷1210=.14, or in other words the saving would be 14 per cent. on all of the fuel used over what it would be if the water were pumped in cold.

In the case of a single boiler using 1½ tons of coal per day, the cost of which is $3.50 per ton, the total saving on fuel for one year of 300 days would be $3.50 \times 1.50 \times .14 \times 300 = \$220$. It is not surprising, therefore, to note the very large number of heaters that are manufactured at the present time, and that the variety of styles offered is so very great. If a heater is not giving satisfactory results an improvement may be made by covering it with some non-conducting covering, but the best results cannot be obtained unless the heater is large enough to allow the water to pass through it at a very slow rate of speed, that it may have time to take up the heat before it is discharged. In testing the heat of feed water, a valve near the heater should be opened, and the water allowed to run on to the bulb of a thermometer.

If the candidate is asked to tell the difference between forced and natural draught he should state that by natural draught is meant the circulation of air through an ordinary chimney, caused by the difference in the weight of air on the outside and the air and gases on the inside. The latter being the lightest are forced upward by the weight of the former, hence the draught is called natural.

If a fan blower or steam jet is used to force the air and gases up through the chimney, the draught is said to be forced. If the gases, or rather the products of combustion which are in the chimney are very hot they will be very light, and if they are very light they will ascend quickly, or in other words, the draught will be very strong. Thus may we reason from cause to effect as far as the draught is concerned, but when we go a little farther we see at once that if these products

of combustion are very hot, it means a loss of heat, which in turn means money thrown away.

Now, if we close our ash-pit doors, and supply the air necessary for combustion by means of a pipe led into the ash pit, which, if supplied by a fan, and we regulate the discharge of air so that we can get just enough to burn the required amount of coal or, in other words, to keep up steam, it is plain that we can create a draught, and a strong one, too, without depending on having a large amount of heat go up the chimney to make it. To get the full advantage of this arrangement, we must have a large amount of heating surface to absorb the heat. From this it will be seen that forced draught is more economical than natural draught is, provided all the conditions are right for it.

The author does not wish to be quoted as being an advocate of using forced draught under the conditions sometimes found where it is used, for he is not. Such conditions are that the natural draught is used as long as it is sufficient to make steam enough to run with, and then the forced draught is substituted, simply because a stronger draught can be secured by so doing. We do not believe in running a plant in this way, as it is not economical in the use of fuel, and it also tends to wear out boilers much faster than when they are used under proper conditions.

In designing chimneys it is a good idea to make the area equal to the combined area of the tubes discharging into it, plus 10 per cent. for a round chimney or 20 per cent. for a square one. The height of it will depend on the surroundings to a great extent. It should not be less than 80 feet, and if there are high buildings in close proximity to it, then it must be carried up higher. Some-

times the area of the chimney is determined by the amount of coal burned on the grates per hour, as follows: First determine the necessary height of chimney. Then multiply the pounds of coal that it is desired to burn per hour by the constant whole number 15. Divide the product so obtained by the square root of the height of the the chimney.

The quotient will be the area of the chimney in square inches. For illustration, suppose that we have decided to erect a chimney 120 feet high, and wish to burn 12 pounds of coal per hour per square foot of grate surface, our grate being six feet square: 6×6=36 square feet of grate; 36×12 =432 pounds of coal per hour; 432×15=6480. The square root of 120 is 10.95, and 6480÷10.95= 591.7 square inches.

By consulting a table we find that a circle 27.5 inches in diameter contains 593.95 square inches, which is near enough for all practical purposes, therefore the diameter of our round chimney or stack should be 27.5 inches. Or 591.7÷.7854= 753.3 and the square root of 753.3 is 27.45, which gives us the diameter practically the same as before.

CHAPTER XXI.

DESCRIPTION OF THE SEVERAL PARTS OF AN INDICATOR DIAGRAM.

In this connection, a general knowledge of the steam engine indicator, its construction, utility, use and care is essential.

The applicant should be able to give readily a description of the principles on which it works, tell the names of the different parts of an indicator card, explain how defects in engines are detected by it, and give full instructions as to how to calculate the mean forward pressure from it, also the back pressure, and explain how to lay out the perfect expansion line, and estimate the water consumption from the card.

The indicator consists of a small cylinder, in which there is a nicely fitted piston made so as to be steam tight without the use of packing rings of metal, and by means of suitable connections this piston operates a pencil, which draws the card on a piece of paper, and to the intelligent engineer this card tells just what is transpiring in the cylinder, so far as variations in the pressure are concerned, as plain as if they were written out by a typewriter.

The indicator card does not tell us everything that we wish to know along this line, for in reality it is but a steamgage, telling us how much pressure we have at all parts of the stroke, taking the line of perfect vacuum as a basis. Its principal

use is to tell whether the valves are set right or not, and to measure the power developed by the engine. If a small card is taken at one time and a larger one at some other time, the difference between the two represents the difference in the load at the time when they were taken, other conditions being the same.

The indicator is of necessity a delicate instrument, and as such needs to be well taken care of, for even slight defects in it lead to conclusions that are erroneous, sometimes seriously so. A full description of all of the parts of a card is not necessary here, but mention of them by way of a reminder may be of interest.

The admission line shows how and when the steam is admitted, the steam line shows how the amount admitted compares with the amount required to get best results, the expansion line helps to locate leaks, the line of exhaust opening tells us when the exhaust valve releases the steam, the counter-pressure line tells us how much back pressure we have, and the compression line tells us when the exhaust valve closes.

After a card consisting of the above lines is taken, the steam is shut off, and another line drawn, which is called the atmospheric line. From this a perfect vacuum line is located by measuring downward, using the scale corresponding to the spring used in the instrument when the card was taken, In places that are near the sea coast or situated on low ground inland, it is customary to put it at 14.7 pounds below the atmospheric line, but the author received a letter a short time ago, from a friend, who lives where the atmospheric pressure is but 11.5 pounds, so that when a card is taken from his engine, the vacuum line must be drawn 3.2 pounds higher up than is

ordinarily done. We know of no inflexible rule for calculating this pressure by the height, for the air grows lighter very fast as we ascend, and

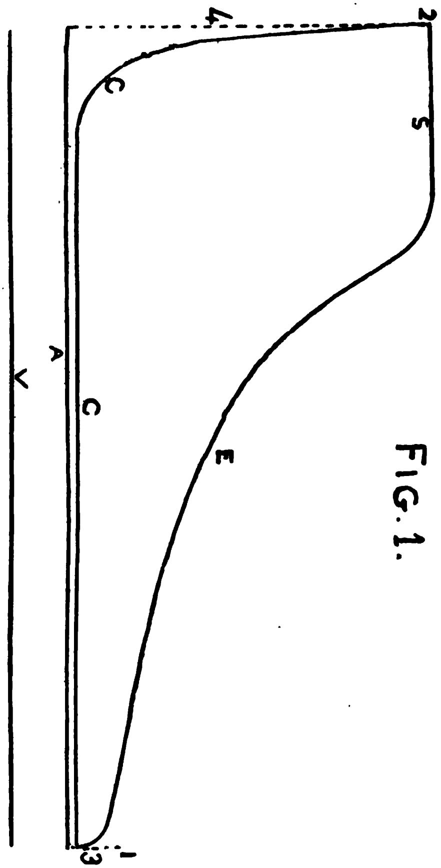


FIG. 1.

moreover, it is not always of the same density or weight in the same place, the variations being indicated by the barometer and the thermometer.

Scientific observations made by a party traveling to the Rocky Mountains show that at an elevation of 6000 feet above sea level, the pressure was 11.48 pounds; when a point a little more than 14,000 feet high was reached it was 7.1 pounds.

What we ordinarily call forward pressure of the steam on the piston is found as illustrated by Fig. 1, in which S is the steam line of a card, E the expansion line, A the atmospheric line and V the line of perfect vacuum. From the point 1 draw a line 3 to the atmospheric one, and at right angles to it. From the point 2 draw a similar one, 4. The space enclosed by 3 on the right, the steam and expansion lines on the top, 4 on the left and A on the bottom represent the forward pressure as we usually speak of it. If we wish to obtain the absolute forward pressure the lines 3 and 4 must be continued down to the vacuum line V, and this line then becomes the boundary on the bottom instead of the line A.

What we usually call the back pressure on the piston is represented by the space enclosed by the line 3 on the right, and the counter pressure and compression lines C C on the top, the line 4 on the left and the line A on the bottom. If we wish to obtain the absolute back pressure, we must take the line V as the boundary of the lower side of the space. The difference between the two, namely, the forward and the back pressure, is the mean effective pressure. There are at least three ways of finding the mean effective pressure from the card, and one of them is to divide the length of it into 10 parts, and ascertain the average height of the upper line above the lower one, measured on the scale corresponding to the spring used in the indicator when the card was taken.

Another way is by using the planimeter to trace over the card, and note the readings of the vernier. Then divide the number of the spring used by the length of the card, and multiply the quotient by the reading of the vernier. The product will be the mean effective pressure. Another way is to begin the tracing with the planimeter, at some point at the right-hand side of the card, and going over it in the direction traveled by the hands of a watch, until the starting point is reached. Then cause the tracer to travel in a perpendicular line until the vernier stands at zero or 0. The distance between the two points measured on the corresponding scale is the mean effective pressure.

CHAPTER XXII.

RULES FOR LAYING OUT THE THEORETICAL EXPANSION LINE.

In Chapter 11 we promised to give a rule for laying out the theoretical expansion line, and now is a good time to do it.

Referring to Fig. 2, which is an ordinary indicator card, we will first draw the vacuum line V at a point 14.7 pounds below the atmospheric line, and the clearance line B at a point far enough from the admission line to represent the percentage of clearance. We will now make a dot on the expansion line at D and from it draw a line parallel to the atmospheric line, represented by D F. Now draw another line from D at right angles to the atmospheric line, as at D G.

Next from the point P draw lines 1, 2, 3, 4, 5, 6 and 7. From the points where these lines cross the line D F erect perpendicular lines 0, 0, 0, 0, 0, 0, and from the points 1, 2, 3, 4, 5, 6 and 7 draw horizontal lines. The points where these lines intersect are the points where the expansion line should be.

For cards where the point of cut-off is short, a very convenient way of laying out the expansion line may be found illustrated in Fig. 3, in which the vacuum line is drawn in its proper place as before, and also the clearance line. Draw the first vertical line through the point of cut-off, and the others the same distance apart that this is from

the clearance line. Starting from the point of cut-off draw lines to the base of the vertical lines as shown. Where the slanting lines intersect the

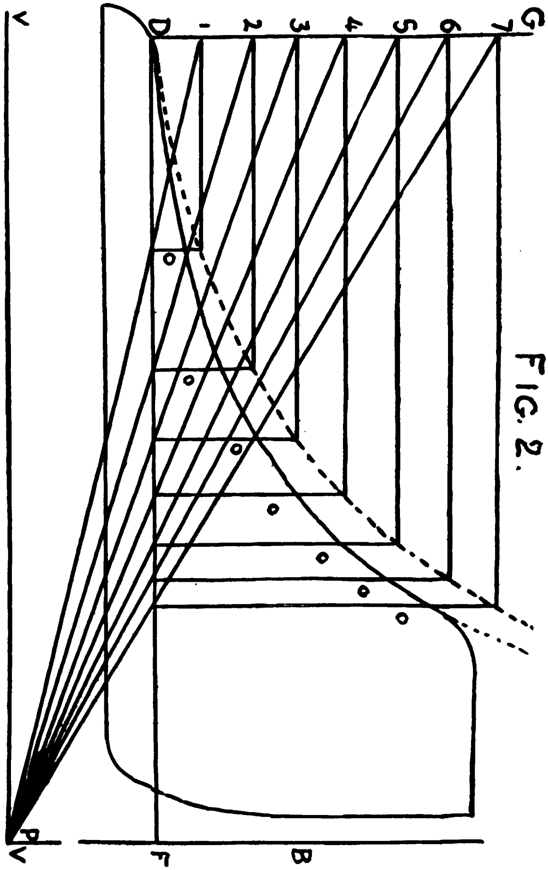


FIG. 2.

vertical ones are the points through which the expansion line must pass, as shown in the figure.

The rule that was given in Chapter 11 for de-

termining the mean effective pressure when the initial pressure and point of cut-off are known is a good one, and this rule it may be remembered

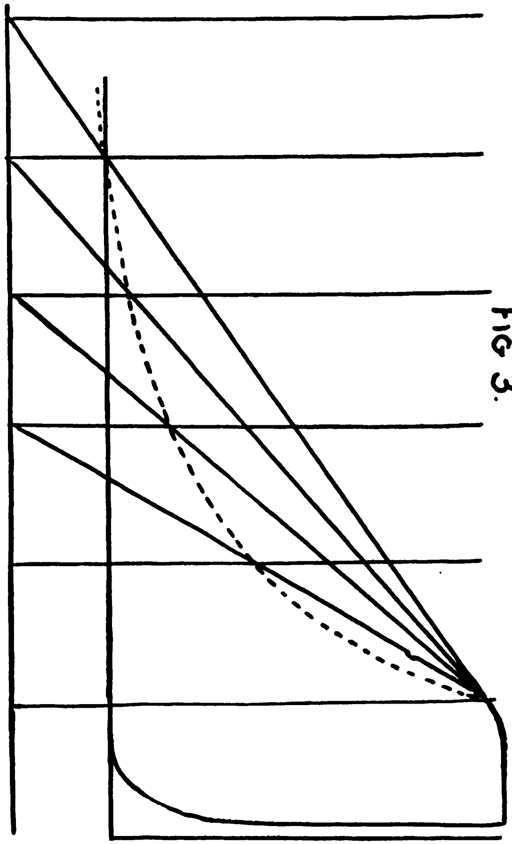


FIG 3.

gave us a mean effective peessure of 41.788 pounds under stated conditions, but wishing to demonstrate the matter to our own satisfaction we con-

structed a theoretically perfect card, but without the clearance line for convenience, assuming the spring to be 40. We then carefully measured off the ordinates, and obtaining the mean effective pressure from them found it to be 42 pounds.

We then went over it with a planimeter and found the reading of the vernier to be 5.16. The length of the card is five inches; $40 \div 5 \times 5.16 = 41.28$ pounds. Going over it again with a planimeter and after returning to our starting point, traveling in a vartical line until the vernier stood at zero, and measuring the distance between the two points, we found that the mean effective pressure was 41.5 pounds, and so we believe that the rule which calculated it to be 41.788 was correct.

It is quite possible that cards may be furnished the candidate for a license which were taken from engines whose valves were improperly set, and he may be required to tell what the trouble was with them, and what the remedy is.

Among other things that are shown by the card, we are told that from it we can calculate the amount of water used by the engine in developing power. This calculation is founded principally on the fact that if we know what the pressure of the steam is at the end of the stroke, we can ascertain what its weight is from tables published for that purpose. As a pound of water evaporated into steam will still weigh a pound, if we know the weight of the steam it is an easy matter to calculate how much water it took to make that steam. But after we have made all our calculation along this line the question naturally arises as to whether this will account for all of the water or not.

A moment's reflection will convince us that it does not Suppose that 10 per cent. of the steam

is condensed upon being admitted to the cylinder. What then? The amount so lost is immediately replaced by more steam from the boiler, the pressure is created and maintained, and as the indicator is simply a steam gage, it can give no intimation of the loss of steam, or rather of water, from this cause. There is also a loss from radiation and other causes, so that when the water consumption is calculated from the card, and then a comparison made with the amount actually pumped into the boilers and they do not agree, there is always some one ready to decry "theory" and claim that practice alone is of value.

Where the water consumption is calculated from the card, it is not always proper to take the actual terminal pressure as showing the amount of steam used, for the exhaust valve may open before the piston reaches the end of its stroke, and so cause the pressure to fall much more rapidly than it would were the fall due to expansion alone. From this it will be seen that the expansion line should be continued to the end of the stroke, and the terminal pressure measured from it, or usually it will be all right to take the pressure at the point of release, as the difference will be very small. Care should be taken to measure this pressure from the line of perfect vacuum. The water consumption may also be calculated from the pressure at the point of cut-off. Rules will be given and explained at length in succeeding chapters.

CHAPTER XXIII.

CALCULATING THE WATER CONSUMPTION OF AN ENGINE FROM AN INDICATOR DIAGRAM.

Our first move in explaining the rule or rules for calculating the water consumption by the card, or the water accounted for by the indicator, will be to draw the card Fig. 4. We have drawn it full size, in order that it may be the more readily understood. We believe that this card was taken from an engine whose cylinder is 24 inches in diameter, and the stroke is 48 inches, and the speed is 65 revolutions per minute. There was a No. 40 spring in the indicator when it was taken. The point 2 is the point of cut-off, 3 is the point of release, 5 the point of compression, V the vacuum line, and 1 and 4 are lines drawn for convenience in making measurements.

A few moments consideration will show that the pressure of the steam, taken for the purpose of calculating the water consumption, must be reckoned from that existing at the point of release 3, or perhaps, more properly speaking, what the pressure would be if the expansion line were extended until it intersected the dotted line 4, which means the end of the card. It will answer every purpose in a majority of cases to take it at point of release, as the difference will be unappreciable.

We repeat, then, that it is proper to take the pressure at the point of release, for it makes no

difference what the initial pressure is, or where the point of cut-off is located, or how much steam was admitted after the cut-off valve closed, for the

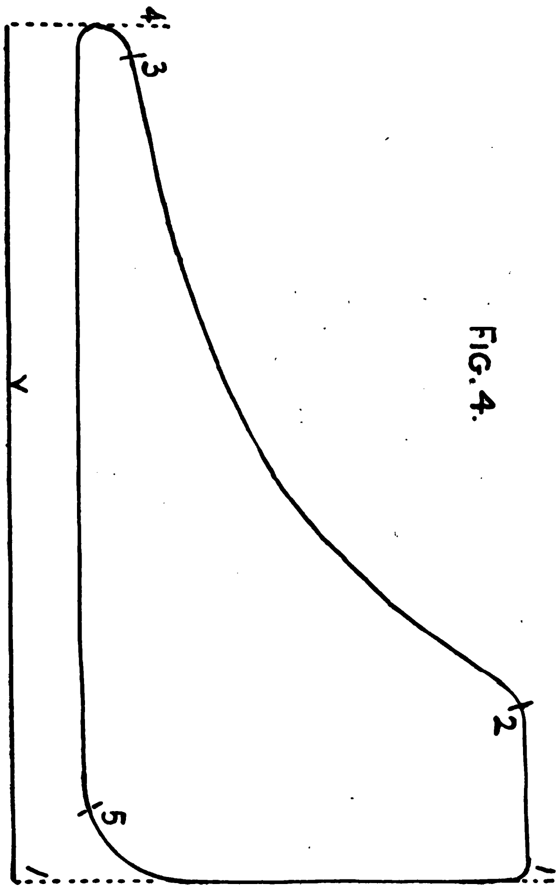


FIG. 4.

pressure existing after all of these operations have taken place is that which indicates the weight of the steam, and from this we must

decide the amount of water required. In this case the absolute pressure at 3 is 27 pounds. Referring to "A Manuel of Rules, Tables and Data for Mechanical Engineers," by D. K. Clark, we find that one cubic foot of this steam weighs .0673 pounds. We must now ascertain how many cubic feet of steam at this pressure we are using per hour, and we shall be able to tell how much the whole of it weighs, and the number of pounds accounted for, per horse power.

From this it may be seen that the cubical contents of the cylinder must be ascertained, but the space occupied by the piston rod will not be taken out, as it will affect the result but very little. The area of a 24-inch circle is 452.39 square inches. The stroke is 48 inches; 452.39×48=21,714.72 cubic inches, which is the volume of the cylinder reckoned by the stroke, but asthe clearance must be filled with steam at each stroke it must be taken into account. It is stated at four per cent. 21,714.72×.04=868.58, and 21,714.72+868.58=22,583.3 cubic inches for each stroke. As the engine runs 65 revolutions each minute, this space must be filled with steam 130 times during each 60 seconds; 22,583.3×130=2,935,829 cubic inches, and by dividing this number by 1728 we have 1699 cubic feet of steam used per minute. As one cubic foot weighs .0673 pound we multiply again and 1699×.0673=114.3427 pounds per minute. As we wish to know the amount per hour we multiply by 60 and our product is 6860.56 pounds per hour. This then is the total amount of water accounted for by the indicator, according to this mode of calculating it. Our next move is to determine the horse power developed.

The horse power constant is 7.13, and the mean effective pressure is 52.4, therefore 7.13×52.4=

373.6-horse power. Dividing the total weight of the water accounted for by the horse power developed we have 6860.56 ÷ 373.6 = 18.35 pounds of water per horse power per hour. This it will be noted is a very fair performance for a non-condensing engine. To some engineers this calculation may seem to be a very elaborate one, requiring many figures to put it in the proper form for inspection, and it is hoped that the matter has been made plain step by step as it has progressed, and while no one can be expected to carry all these figures in his head, and draw them out to order when wanted, still if any one who is not thoroughly familiar with the manner of solving such problems will study out the principle involved it will make what appeared to be a hard lesson to start with an easy one at the finish, and such calculations have an attraction for the engineer who is thoroughly interested in his business, as they tend to relieve the monotony of the ceaseless routine of work in the engine and boiler-room, and furthermore the author will say, while speaking for himself, that these matters possess a fascination which is equalled by few others and excelled by none. But to return to the subject. However lengthy the above calculation may seem it is incomplete, as it does not take into consideration all of the factors which form a part of the complete whole. It will answer for all practical purposes, and is by no means to be despised, but a formula will now be referred to which embraces every point that can effect results, a most thorough and complete rule, and in fact, it may be termed a masterpiece.

CHAPTER XXIV.

MORE ABOUT CALCULATING THE WATER CONSUMPTION OF AN ENGINE.

We will first give the formula as we find it for determining the water accounted for per horse power per hour at the point of release. This formula is said to be used in the Massachusetts Institute of Technology:

$$\frac{13750}{\text{M.E.P.}} \overline{(R+E\ Wr)-(H+E\ Wh)} =$$

number of pounds of steam accounted for at release, in which, together with another formula using pressure at point of cut-off, the following symbols are made use of: M E P= mean effective pressure; C=portion of stroke completed at cut-off expressed in decimals; E=per cent. of clearance or the volume of clearance compared with the volume of the cylinder; R=proportion of stroke completed at release; H=proportion of stroke uncompleted at compression; Wc=weight of one cubic foot of steam at cut-off pressure; Wh=weight of one cubic foot of steam at compression pressure; Wr=weight of one cubic foot of steam at release pressure.

An explanation of this formula is as follows: The constant whole number 13,750 is to be divided by the mean effective pressure, and the quotient so obtained we will set down by itself, calling it our first answer. To R we must add E, and mul-

tiply their sum by Wr, and the product so obtained we will call our second answer. To H we must now add E, and multiply their sum by Wh, and this product we will call our third answer for convenience. Subtract the third from the second and multiply the remainder by the first. The product will be the pounds of water per horse power, per hour, at release pressure.

We will now work out an example by this formula, taking for illustration the card shown in Fig. 4, Chapter 23, and when we substitute for the symbols their values taken from this card, our formula is as follows:

$$\frac{13750}{52.4}\,(.964+.04\times.0673)-(.071+.04\times.0411)=16.53$$

pounds of water per horse power per hour. This is somewhat less than the result obtained by a former rule, but from the conditions stated it is only reasonable to suppose that it would be from the start, for the simple reason that while the first rule given counts all the steam lost, when it is released by the exhaust valves, this rule takes into account the fact that a portion of it is saved by the closing of the exhaust valve before the completion of the return stroke, in other words by compression.

In order to avoid any misunderstanding on the part of those who are not familiar with such computations as these, we shall now offer a full explanation of the way in which the figures were obtained. The total length of the card is 5.25 inches, and by measuring from the end of the card to the point 3, we find that 27-28ths of the stroke has been completed when the exhaust valve opens, and reducing it to a decimal fraction, we have .964. That is how we obtained R. The

clearance is estimated at 4 per cent. of the volume of the cylinder, judging by the distance from the piston to cylinder head when on the centre, plus the volume of the ports, etc. Thus we obtained E. Wr was obtained from a table in the manual referred to previously by D. K. Clark. By measuring from the point 5 to the end of the card, we find it to be three-eighths of an inch, or one-fourteenth of the whole stroke. 1-14=.071. In this way we obtained H. Wh was obtained from Clark's manual. If it is desired to obtain the water accounted for by the indicator at the point of cut-off, the following formula should be used:

$$\frac{13750}{\text{M.E.P.}}\,\overline{(C+E\,W_c)-(H+E\,W_h)}=$$

pounds of water per horse power per hour. When we substitute for the symbols their values as determined by the card Fig. 4, we have the following:

$$\frac{13750}{52.4}\,\overline{(.25+.04\times.2586)-(.071+.04\times.0411)}=18.48$$

pounds per horse power per hour at the point of cut-off.

The process is as follows: 13750÷52.4=262.4, which is our first answer. .25+.04×.2586=.074994, which is our second answer. .071+.04×.0411=.004562. .074994—.004562×262.4=18.48 pounds. In this case the author has pursued his usual policy of illustrating several rules given, by using the same data for all.

It is a policy not always looked on with favor, as it shows up the difference too well, but as we are seeking after knowledge, and as this book is written for the purpose of imparting the same, it is better to look at these matters in their true

light. When two or more rules are given by good authority it is often a difficult task to decide which one is correct, but if the candidate is well versed in his favorite, and can explain it in detail, probably no inspector would refuse him a license on account of small differences in the results obtained. Some further comparison will be made in the next chapter.

CHAPTER XXV.

STILL MORE ABOUT CALCULATING THE WATER CONSUMPTION.

The following formula may be used for calculating the water accounted for at the point of cut-off:

$$\frac{V\,S\,W}{H.\,P.} = P$$

In which V=volume of cylinder up to point of cut-off in cubic feet, S=number of strokes per hour, W=weight of one cubic foot at cut-off pressure (absolute), H. P.=the horse power developed, and P=the pounds of water per horse power.

Assuming the elements of the preceding case, and again referring to Fig. 4, Chapter 23, we find that the area of the cylinder is 452.39 square inches. The cut-off takes place at 12 inches, and $452.39 \times 12 \div 1728 = 3.14$ cubic feet. The speed is 65 revolutions per minute, or 7800 strokes per hour. The pressure at cut-off is 113 pounds, and the weight of one cubic foot is .2586 pound. It was developing 373.6 horse power.

Then $3.14 \times 7800 \times .2586 \div 373.6 = 16.95$ pounds per horse power per hour.

It will be noticed that we have not taken the clearance into account so far, for the reason that we wish to raise the question as to whether it should be a factor in the calculation, inasmuch as

it is filled by compressing the exhaust steam. As it is generally conceded that the amount of clearance in the cylinder will effect the economy of the machine, and some authorities at least deem it proper to recognize it when making calculations, we should like to present another question concerning it. In this case it is four per cent., that is, it possesses .04 of the volume of the whole cylinder, but if we are using the volume of it up to the point of cut-off only, is it proper to call it the same? As we have reduced the entire volume under consideration to one fourth of what it was, and have not reduced the volume of the clearance, is it not proper to say that the percentage is four times as much, or 16 per cent?

Viewing it in this light then we have 16.95 pounds per hour; adding 16 per cent to it makes it 19.66 per horse power per hour. Putting the results of the four rules together we have for the

1st	rule,	taking	pressure	at	release,	18.36	pounds
2d	"	"	"	"	"	16.53	"
3d	"	"	"	"	cut-off	18.48	"
4th	"	"	"	"	"	19.66	"

Of course rules No. 1 and 2 should be classed together, as the conditions are the same, and the difference in the results are not so large as they at first appear to be, for it is but 82 gallons per hour, even for this large engine, or less than 22 gallons for each 100 horse power developed. Rules No. 3 and 4 may be compared, and the difference between the results is but 53 gallons per hour, or less than 15 gallons for each 100 horse power developed.

In the rules No. 2 and 3 it will be noted that the water consumption is calculated directly from the card, without bringing the size of the cylinder into the calculation at all. This can be done

where the rate only is required, as in this case, and shortens up the process materially.

We wish to emphasize this, for the answer so

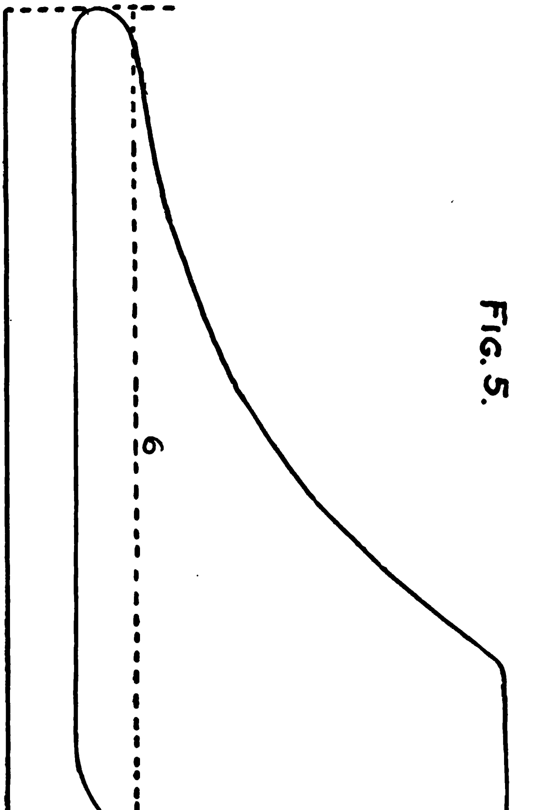


FIG. 5.

obtained is the amount of water accounted for per one horse power per hour, and if the whole amount accounted for is desired (when employing

rules 2 and 3), it must be multiplied by the power developed. In this book it has been one of the objects of the author to dispense with the use of tables as far as possible, but there is one which makes the calculation of the amount of water accounted for by the indicator such a simple matter that it is worthy of notice.

This table was originally prepared by Mr. E. W. Thompson for the American Machinist, and in its original form is very full and complete, embracing the terminal pressures from three to 60 pounds absolute, both included, advancing by tenths of a pound, but for all practical purposes the whole and half pounds are sufficient. To use the table, first ascertain the absolute terminal pressure and take the number opposite to it in the table, which is to be divided by the mean effective pressure. The quotient so obtained will be the pounds of dry steam accounted for per horse power per hour. For the purpose of illustrating the utility of this table, let us refer to Fig. 5, which is a reproduction of a card shown in Fig. 4. The pressure at the point of release is 27 pounds absolute. The number in the table which is opposite 27 is 927.990. The mean effective pressure of this card is 52.4 pounds; $927.990 \div 52.4 = 17.71$ pounds dry steam per horse power per hour, or in other words, water accounted for by the indicator.

This as it stands is uncorrected for clearance and compression. If it is desired to do this, draw the dotted line, 6, in the figure, through the point of release parallel to the atmospheric line, and also the two vertical lines representing the extreme ends of the card. Ascertain the distance from the end of the card at the left to the place where the dotted line crosses the compression

line at 7. Multiply the pounds of water already accounted for by this distance in inches, and divide the product by the distance between the two vertical dotted lines. The quotient will be

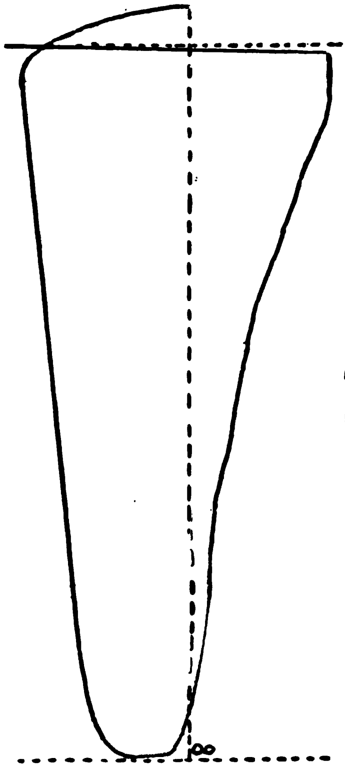


FIG. 6.

the water accounted for per horse power per hour, corrected for clearance and compression. Applying the above we find that 17.71×5.1875 = 91.8706, and dividing this by 5.25, we find our

quotient is 17.50 pounds. In many cases found in practice, however, the conditions will be somewhat different than with this card, an illustration of them being shown in Fig. 6. In this case the terminal exceeds the compression pressure (both being absolute); therefore it will be necessary to extend the compression line as shown and proceed as before.

The data in this case is as follows: Release pressure 28 pounds; number opposite 28 in table is 960.120, and the mean effective pressure is 19 pounds; 960.120÷19=50.53 pounds per horse power per hour. To correet for clearance and compression: The distance from the end of card at 8 to where the dotted line crosses the compression line is 3⅞ or 3.875 inches, and from the same point to other end of card is 3 11-16 or 3.6875 inches; 50.53×3.875=195.8037 and dividing this by 3.6875 gives us a quotient of 53.1 pounds per horse power per hour. It will be noted that this amount exceeds that obtained at first, which is what might be expected, as the compression is less than the terminal pressure.

T. P.	Number.	T. P.	Number.
3	117.300	3.5	135.748
4	153.880	4.5	171.945
5	189.750	5.5	207.598
6	225.240	6.5	242.970
7	260.540	7.5	278.063
8	295.440	8.5	312.800
9	330.030	9.5	347.273
10	364.400	10.5	381.570
11	398.640	11.5	415.725
12	432.720	12.5	449.688
13	466.570	13.5	483.435
14	500.220	14.5	517.070
15	533.850	15.5	550.638

T. P.	Number.	T. P.	Number.
16	567.360	16.5	584.100
17	600.780	17.5	617.400
18	633.960	18.5	650.460
19	666.900	19.5	683.378
20	699.800	20.5	716.270
21	732.690	21.5	749.060
22	765.380	22.5	781.763
23	798.100	23.5	814.393
24	830.640	24.5	846 965
25	863.250	25.5	879.495
26	895.700	26.5	911.865
27	927.990	27.5	944.075
28	960.120	28.5	976.268
29	992.380	29.5	1008.458
30	1024.500	30.5	1040.508
31	1056.480	31.5	1072.418
32	1088.320	32.5	1104.350
33	1120.350	33.5	1136.420
34	1152.260	34.5	1168.170
35	1184.050	35.5	1199.900
36	1215.720	36.5	1231.693
37	1247.640	37.5	1263.563
38	1279.460	38.5	1295.333
39	1311.180	39.5	1327.003
40	1342.800	40.5	1358.573
41	1374.320	41.5	1390.043
42	1405.740	42.5	1421.413
43	1437.060	43.5	1452.900
44	1468.720	44.5	1484.520
45	1500.300	45.5	1516.060
46	1531.800	46.5	1547.520
47	1563.220	47.5	1578.900
48	1594.560	48.5	1610.200
49	1625.820	49.5	1641.420
50	1657.000	50.5	1672.560
51	1688.100	51.5	1703.620
52	1719.120	52.5	1734.600
53	1750.060	53.5	1765.500
54	1780.920	54.5	1796.320

T. P.	Number.	T. P.	Number.
55	1811.700	55.5	1827.338
56	1842.960	56.5	1858.568
57	1874.160	57.5	1889.738
58	1905.300	58.5	1920.848
59	1936.380	59.5	1951.898
60	1967.400	60.5	1982.888

CHAPTER XXVI.

CONCLUSION OF THE SUBJECT OF WATER CONSUMPTION.

There is still another style of card to which we wish to apply the rule, which makes use of a table, an illustration being shown in Fig. 7. In this case the cut-off is very short, and as a consequence the terminal, or, more properly speaking, the release pressure, is below the atmosphere. This makes no difference, so far as determining the water accounted for is concerned, but inasmuch as it may prove a stumbling-block to some reader, we have deemed it best to introduce it at this point. Where the steam is expanded below the pressure of the atmosphere, and the exhaust valve is opened before the end of the stroke, the pressure rises instead of falling, or, in other words, is just the reverse of what takes place when the release pressure is above the atmosphere, therefore, the rule which tells us to continue the expansion line to the end of the card, carrying it out in the same manner that it would have been carried out, provided the exhaust valve had not opened before the completion of the stroke, applies to this case as well as to others which have been illustrated. As the release and terminal pressure are so much below the compression, we must continue the compression line downward as shown at 9 in the figure. Applying the rule we find that the terminal pressure is 10.5 pounds absolute. The number

opposite 10.5 in the table is 381.57. The mean effective pressure is 10.25 pounds; 381.57÷10.25 = 37.22 pounds of water accounted for per horse

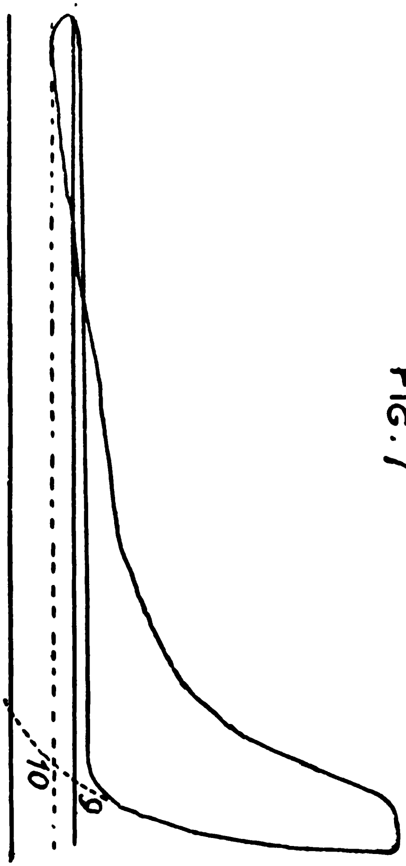


FIG. 7

power per hour uncorrected for clearance and compression.

The distance from the end of the diagram at the

left to the point where the two lines cross at 10 is 3.875 inches, and the total length of the card is 4.25 inches; $37.22 \times 3.875 \div 4.25 = 33.93$ pounds corrected for clearance and compression. The engine from which this card was taken is one of the best made, and its builder has a world-wide reputation as a builder of first class machines, and yet the water rate is high, and its efficiency low accordingly, but what is the cause of it? Simply because it is underloaded. Suppose that more machinery should be put into the factory which is run by this engine until the terminal pressure would be 25 pounds instead of 10.5 pounds as at present. Then the water consumption would be reduced to about 20 pounds, although no change had been made in the engine. This shows the loss due to running an underloaded engine, but at the same time the machines in that factory are all running up to full speed, which could not be said if the engine had a heavy load to carry, for there would be times when the speed would be reduced, causing the output to be lessened, which would be a greater loss than is caused by what fuel is wasted under the present conditions, and this is a leak that cannot be detected by the indicator.

Concerning a rule for calculating the water rate in the case of a compound engine, would say that we know of no better one than rule No. 1 as already given for a simple engine, but in using it the high-pressure card should be made use of, and care taken to divide the total amount of water accounted for by the total power developed in both cylinders.

The first rule mentioned, in which the table given in Chapter 25 is used, may be applied to the case of a compound engine, using the diagram

from the low pressure cylinder for the calculation.

These rules are useful for the purpose of making comparisons of different engines, and also of the performance for the same engine under varying conditions.

CHAPTER XXVII.

CAUSES FOR DEFECTIVE INDICATOR DIAGRAMS.

While the indicator points out many defects in valve setting, and also in the construction of the steam engine, still when a card is taken and found to be imperfect, it is not wise to "jump to a conclusion" as to the cause of the trouble, as there may be more than one cause for it. Take for instance, the case of a card or diagram that shows a higher terminal pressure than should be according to the ways of laying out the theoretical expansion line illustrated in Chapter 22. We would naturally conclude that the steam valve leaked, thus admitting more steam after the cut-off had taken place, and raising the expansion line accordingly, but such a conclusion might be an erroneous one, as the defect may be due to re-evaporation, that is, there may be water in the steam when first admitted to the cylinder, or it may be due to initial cylinder condensation, by which we mean that the cylinder walls may have been cooled down by a low terminal pressure during the previous stroke, so that when steam is again admitted, although it may have been dry, still the effect of this lower temperature is to cause some of the incoming steam to be condensed at once, but after the cut-off has taken place and the pressure grows less as the piston advances, this water of condensation is again evaporated by the heat present, and hence the expansion line is raised.

Again, it is often inconvent to ascertain the exact clearance, so that the best that we can do is to approximate it, and if it is really more than we have calculated on, then the actual expansion line will be above the theoretical one, although no steam has been admitted since the cut-off took place. On the contrary, if the actual is below the theoretical expansion line, it may be due to the fact that, in estimating the volume of the clearance, an error has been made in calling it more than it really is, so that the only cause for difference between the two lines is this error in the beginning of the process. However, if the engineer is satisfied that no error exists here, the fall in the expansion line points to a leak in the piston, for, as it is pushed forward, some of the steam passes by it, over into the other part of the cylinder hence the undue fall in pressure. Or it may be that the exhaust valve leaks, thus allowing some of the steam to pass out to the exhaust pipe without doing its share of the work.

There is at least one other reason for this fall in the line which will only apply to extreme cases, and that is excessive condensation, after a cut-off has taken place. If an engine is located in a very cold place, without suitable protection by lagging or covering with some non-conductor, and the weather is cold, then the condensation is great, and will cause the line to fall when no defect exists in the engine itself. If the line falls in a proper manner for a time and then suddenly rises, it shows that the steam valve is reopened, thus admitting another charge of steam, before the completion of the stroke. In the case of some engines it is not such an easy matter to locate the cause of such distortion as might be supposed. A certain engine having valves which are

actuated in a similar manner to the Corliss valve, was found to be in a bad condition. A diagram taken from it showed that the steam valve reopened at about five-eighths of the stroke. An examination of the eccentric showed that it was in its proper place. On the hub of the jim-cranks which operated the steam valve was a mark which corresponded to a similar mark on a collar located on a sleeve which formed a bearing for the valve stem. With the engine on the centre, these marks were found to be in such a position as to form one continuous line, thus showing that the valve was properly set. Still, when another diagram was taken, it was very plain that the valve reopened as before. When the bonnet on the other side of the cylinder was taken off to compare the marks on valve and cylinder, it was found that they did not agree, thus showing plainly that the valve was not properly set, and at the same time pointing out the fact that it was due to a twist in the valve stem.

If the expansion line falls very suddenly at say three-quarters stroke, it indicates that the exhaust valve opens sooner than than it ought to. If the counter-pressure line rises suddenly before it is time for compression to begin, it indicates that the exhaust valve is closed sooner than it ought to be, and if the compression exceeds the initial pressure, it also shows that the exhaust valve closes too soon, and this is not only a wasteful condition of affairs, but a dangerous one as well. It is wasteful because it causes the point of cut-off to be lengthened to run the same amount of machinery, causing a direct waste to that extent, and it is dangerous because it brings a greater pressure to bear on the cylinder head than has been calculated on in some cases, and

also because if there should be water carried over for any cause, and the cylinder is not perfectly drained during the return stroke, there is more reason to believe that the effect will be disastrous than if the compression is light, under normal conditions. No inflexible rule can be given for the amount of compression that will give best results, but it is only reasonable to assume that the greater the speed the greater will be the compression advisable, other conditions being equal.

CHAPTER XXVIII.

POWER DEVELOPED BY DIRECT STEAM AND BY EXPANSION.

The indicator diagram enables us to tell, in addition to other things, how much of the work done is furnished by steam direct from the boiler, and how much is furnished by the expansion of this steam.

Suppose that we take an indicator card in which the admission line is at the left hand and the expansion line at the right hand. We will draw a perpendicular line from the point of cut-off down to the line of perfect vacuum. Now all of the space at the left of our perpendicular line between the steam line and the vacuum line represents the work by direct steam pressure, and all of the space at the right of the perpendicular line between the expansion line and the vacuum line represents the work done by the expansion of the steam.

It has been claimed that when we take into account the work done by direct steam pressure we should stop there, as that is all that there is to it, and that to figure in the expansion of the steam is to claim to get double duty out of it, but we do not see how it is possible for this claim to be made good, for when the steam is cut off in the cylinder it is not all used up at once, for the force in it at the point of cut-off has been supplied directly from the boiler, as none of the benefits of

the expansive qualities of it have been made use of, so far as performing work in the cylinder is concerned, and if it were immediately exhausted at this point there is no doubt that it would require more coal to devolop 100-horse power than it does at present.

The proportion of the work done during expansion, to that done during the time that the steam is admitted to the cylinder up to the point of cut-off, is represented by the hyperbolic logarithm of the ratio of expansion. Suppose that the ratio of expansion is four, the hyperbolic logarithm of which is 1.3863. Then if we call the amount of work done previous to cut-off unity, or in other words call it one, then the work done during expansion will be 1.3863, hence the rule. To find the work done during expansion, multiply the hyperbolic logarithm of the ratio of expansion by the work done during the period of full steam. The product will be the work done during expansion. The words "power developed" may be inserted instead of the words "work done," if so desired, for although the terms are not interchangeable as a rule, still it must be remembered that this is an example in proportion only.

It should not be forgotten that such rules do not always take account of clearance or compression, and that the pressures are all absolute, as is always the case when the full power developed by the engine is to be calculated.

To demonstrate the correctness of this rule may seem to be a complicated operation to some engineers, but in reality it is a very simple one. It may be done as follows:

Draw a theoretically perfect card, without clearance or compression, putting the cut-off at one-quarter stroke, making the initial pressure 80

pounds above the atmosphere, or 95 pounds absolute. Now draw a perpendicular line from the point of cut-of to the vacuum line. Having done this, the author took a planimeter and carefully measured the area of the space representing the work done prior to the cut-off point and found it to be 2.96 square inches. Now, as the cut-off took place at one-quarter stroke, the ratio of expansion is 4, the hyperbolic logarithm of which is 1.3863. By multiplying 2.96 by 1.3863, we find that the product is 4.10, which should be the area of the space representing the work done by expansion. As the actual reading of the plainimeter was 4.14, it proves that the rule is correct, for we do not expect that these readings will always agree exactly, and a difference of .04 in a case like this is not of enough importance to be taken into account.

Some writers tell us that if we are running an automatic engine, we should carry such a pressure on our boilers that the steam in the cylinder will expand down to atmospheric pressure at the end of the stroke. Others claim that it is better to have about five pounds above the atmosphere. Probably either one will give good results within reasonable limits, but with a very light load, if we should attempt to lower the boiler pressure to meet this requirement, it would be so low that it would prove to be unprofitable, for although when we expand the steam below the pressure of the atmosphere the condensation is excessive, still it proves to be a less evil than to run with a low boiler pressure. If we were running with a gage pressure of 75 pounds on the boilers, and the load is such as to give a high terminal pressure, say 35 pounds absolute, if we raise the boiler pressure to 100 pounds, the expense for fuel will

be much less, taking it for granted that the boilers are safe at the higher pressure. In the case of a compound condensing engine there are other matters to take into consideration, which do not apply to a simple engine.

CHAPTER XXIX.

SIMPLE AND COMPOUND ENGINES.

If we are running a simple, non-condensing, automatic engine, and for any cause wish to raise the terminal pressure, all that we have to do is to lower the boiler pressure, and the desired result is obtained, and that is all that there is to it; but if we have a compound condensing engine and the terminal pressure in the high pressure cylinder is too low to give good results, there are several things to be taken into consideration. Let us follow the operation and note the results. If the terminal pressure in the high pressure cylinder is very low, then but little steam will be left to go to the low pressure cylinder, and an uneven distribution of the load is a natural consequence. Now if we can raise this terminal pressure it will make a difference, and so we proceed to lower the boiler pressure. Assuming that the load is a constant one, the first thing that we notice is that the point of cut-off is lengthened, and the steam is not expanded down as low as it was formerly. We now have more steam going to the low pressure cylinder, and its mean effective pressure is raised accordingly, but as the mean effective pressure of the two cylinders taken together is a constant factor under conditions named, this increase is not needed, and the effect of it is to cause the engine to make an effort to increase its speed, but this effort is promptly checked by the governor,

which shortens the point of cut-off, the effect of which is to reduce the mean effective pressure in the high pressure cylinder, and also in the low pressure, and as our terminal pressure in this cylinder is now raised, we shall need more water to condense the steam, as a higher pressure means more heat to dispose of, and thus more work is put upon the air pump. In addition to this, as even now the point of cut-off is longer than it was before the boiler pressure was reduced, more steam will be needed on this end of the machine also. Now the question is, have we gained enough to offset the loss?

A due consideration of these facts will explain the necessity of having an engine of this kind properly proportioned for the work that it will be required to do, and for the power that it will be called upon to develop. Of course it is always well to have even a simple engine of a suitable size, but as we add on cylinders the desirability of it increases.

Another question that the indicator helps us to decide, in the case of a compound engine, is whether it pays to run a compound engine or not. Suppose that we have a simple, non-condensing engine, in which there is no back pressure above the atmosphere. The conditions are such that the horse power constant is 2.5 Now if we compound the engine, that is, add another cylinder, say twice the diameter of the first one, we may have caused a back pressure of six pounds in the high pressure cylinder. This will of necessity cause the mean forward pressure to be increased by six pounds, and consequently this cylinder will be called upon to furnish $2.5 \times 6 = 15$-horse power more than it did before, and this is required to overcome the increased back pressure.

If this were not put to some use, the compounding would be a dead loss, but as the extra steam so used is to be utilized further, it changes the the conditions. As the low pressure cylinder is twice the diameter of the high pressure, it follows that the horse power constant will be four times as great, ignoring the areas of the piston rods, so that $2.5 \times 4 = 10$.

Now, if by creating an additional back pressure of six pounds, calling for 15-horse power to overcome it, we can secure steam enough to run our low pressure cylinder, securing for it a mean effective pressure of three pounds, then it shows on the face of it that it pays to compound, for when we trade off 15 horse power and get 30 in return for it, we are on the safe side, to say the least. It is assumed that the extra water needed will not have to be bought of the local corporation monopoly, provided we wish to add a condenser to this engine.

The indicator does not tell us everything that we wish to know in this connection, however, for it does not point out the difference in cylinder condensation, which is often referred to as X on account of its being an unknown quantity, but as the terminal pressure in the high pressure cylinder is higher, it is only reasonable to suppose that the condensation will be less in proportion as the difference between the initial and the terminal pressure is less.

If we now add a condenser, and thereby remove a large part of the back pressure on the large piston, we shall make another improvement, for the air pump deals with this atmospheric pressure in a more economical way than the piston can, therefore it is good policy to make the second cylinder much larger than the first, the ordinary

limit being twice the diameter, or four times the area, but a more common practice is to make them as 16 is to 30 inches, or as 22 is to 40 inches, respectively, in diameter. For cross compounds it is customary to make the piston rods of the same diameter, notwithstanding the great difference in the size of the pistons, on account of the difference in the mean effective pressures acting on them.

CHAPTER XXX.

CALCULATING THE AMOUNT OF WATER NECESSARY TO SUPPLY A SURFACE CONDENSER.

Some time ago a friend in another state requested the writer to inform him how much water he would need, provided he added a condenser to his engine. He stated that it was a compound engine, gave the diameters of the cylinders, length of the stroke and pressure carried on the boiler. As the point of cut-off was not given, nor any data from which the terminal pressure could be calculated, it was not possible to give an intelligent answer. This then is another matter that the indicator helps us to decide, namely, the volume of circulating water that will be required for any engine under stated conditions.

This problem we intend to illustrate and explain in full, beginning with a formula given us for this purpose:

$$\frac{\overline{1114+.3\,T}-t}{t'-t''}=V.$$

In which T=temperature of steam entering condenser, t=temperature of feed water, t'=temperature of water of condensation discharged, and t''=temperature of circulating water, all in degrees F., and V=volume of water.

This formula may be read as follows: To the constant number 1114 add .3 of the temperature

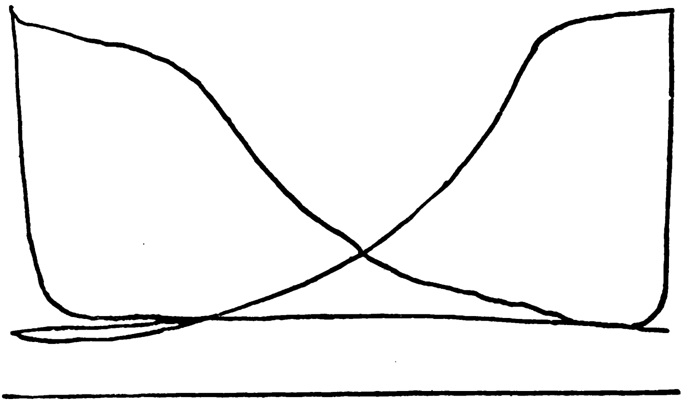
of the steam as it enters the condenser (expressed in degrees Fahrenheit). From the sum so obtained, subtract the temperature of the feed water. Divide the remainder by the temperature of the water of condensation as discharged, minus the temperature of the circulating water before it is taken up by the circulating pump. The quotient will be the volume required; or, in other words, it will be the number by which you must multiply the number of pounds that is required per hour to generate the steam that is used in operating the engine. The product will be the nnmber of pounds needed per hour to supply the condenser.

In calculating the amount of water that will be needed it is well to remember that while it is a good idea to continue to pump the feed water through a heater, still it must not be expected that it will be as hot as formerly, as the pressure in the exhaust pipe will be greatly reduced by the condenser and air pump. Also that after the condenser has been added the back pressure will be reduced, thus requiring less forward pressure, the consequence being that the terminal pressure will be less (as before stated), making the temperature of the exhaust steam as it goes to the condenser lower than when it is exhausted into the atmosphere. Therefore we may safely assume a lower temperature for it, or if we choose to take it as we find it, whatever error there is will be on the safe side, as the result so obtained will be more than what is actually needed, but the difference will not be great.

We now call attention to Fig. 8, which consists of diagrams from the high pressure cylinder of a compound condensing engine, the data being as follows: Diameter of cylinder 12 3-16 inches,

stroke 36 inches. Diameter of piston rod 2⅛ inches. Net area of piston, obtained by subtracting one-half area of rod from area of piston 114.9033 square inches. Horse power constant

FIG. 8.



1.5041. Mean effective pressure of right hand card, 20 pounds, left hand card 23 pounds, and of both together 21.5 pounds. Indicated horse power 32.338, terminal pressure (absolute), right hand card 25, left hand card 28 pounds, the average being 26.5 pounds. Fig. 9 consists of cards from the low pressure cylinder of the same engine, the data being as follows: Diameter of cylinder 20 inches, stroke 36 inches, diameter of rod 3 inches, net area of piston 310.6257 square inches, horse power constant 4.0663, mean effective pressure, right hand 10.5 pounds, left hand 10 pounds, and of both together 10.25 pounds. Indicated horse power 41.679, terminal pressure of right hand 7.5 pounds, of left hand eight pounds, the average being 7.75 pounds.

In determining the horse power developed by this or any other compound or compound condensing engine, all that is necessary is to add together the horse power developed in the two cylinders. In this case it is 32.338+41.679=74.017

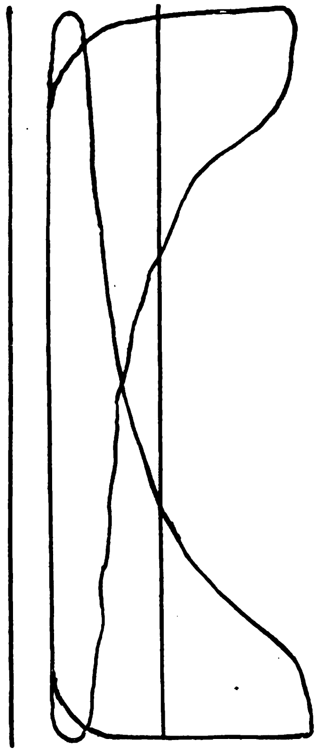


FIG. 9.

horse power. The initial pressure is but 80 pounds. As the lowest terminal pressure is 7.5 pounds, we will assume that the feed-water is heated to 130°, that the steam as it enters the condenser has a temperature of 180°, the temperature

of the water of condensation discharged is 100°, and that of the circulating water 70°. Our problem, therefore, is as follows:

$$\frac{\overline{1114}+.3\times180-130}{100-70}=34.6 \text{ volumes.}$$

We would call attention to the fact that if the temperature of the discharged water is 110°, the value of V will be 25.95, instead of 34.6 as above, which will make a great difference in the work of the circulating pump, and will be quite an object when the supply of water is limited. The next step is to ascertain the amount of water that this engine calls for per hour, which we will do by one of the rules already explained, and that may be found in Chapter 23.

The high pressure cylinder has an area of 116.6766 square inches, and its length is 36 inches. 116.6766×36=4200.3576. Add five per cent. for clearance and we have 4200.3576+210.0178=4410.-3754 cubic inches. Dividing this by 1728 gives us 2.5523 cubic feet per stroke, and as this engine runs 72 revolutions per minute, we have 144 strokes to calculate on, and 2.5523×144=367.5312 cubic feet per minute.

The average terminal pressure of these cards is 26.5 pounds. The weight of one cubic foot of steam at 26 pounds pressure is .0650 pound, and at 27 pounds it is .0673 pound, therefore .0650+.0673÷2=.06615 pound, which is the weight at 26.5 pounds pressure; 367.5312×.06615=24.3121 pounds per minute, and multiplying this by 60 gives us 1458.726 pounds per hour. One gallon of water weights 8.3 pounds, and 1458.726÷8.3=175.75 gallons per hour, which is the water accounted for by the indicator.

Referring to a preceding part of this calculation, we find that we need 34.6 volumes, and 175.75×34.6=6080.95 gallons per hour. It may be of interest to note the efficiency of this engine, and as the indicator accounts for 1458.726 pounds of steam per hour, and as the total horse power developed is 74.017, we divide the former by the latter, and our quotient is 19.7 pounds of steam per horse power per hour.

If this were a larger engine, using steam at say 130 pounds pressure, and with a load for which it is adapted, we should expect a greater efficiency; or, in other words, less steam per horse power per hour would be used, but when we consider the size of it, the low initial pressure of 80 pounds and the slow speed, we admit that it is doing very well.

It was formerly a simple engine, but as the load was increased the low pressure cylinder was added and it is now giving good satisfaction, using less fuel than before, and doing the work easily.

CHAPTER XXXI.

THE AMOUNT OF WATER NECESSARY FOR A JET CONDENSER.—DENSITY OF STEAM.

As will be noted the formula given in Chapter 30 applies to the use of a surface condenser, and it will no doubt be of interest to explain the formula given us for use in the case of a jet condenser, as many of them are in use, and still more will be used in the future.

We shall first give a formula to determine the amount of water that will be required to reduce steam of any given pressure to water of a stated temperature, as follows:

$$\frac{S+L-D}{D-I}=V$$

In which S=sensible heat of the steam, L=latent heat of steam, D=temperature of discharged water, I=temperature of injection water, and V= volume of water as compared with that which it required to generate the steam.

When written out in full this will read as follows: Add the sensible and latent heat of the steam expressed in degrees F. together, and from the sum so obtained subtract the temperature of the discharged water. Divide the remainder by the number obtained by subtracting the temperature of the injection water from the temperature of the discharged water, and the quotient will be the volume required. We will now apply this to

the case mentioned, and illustrated in Chapter 30, Figs. 8 and 9, but we must take the steam as it leaves the low pressure cylinder for the condenser, as that is what we must condense. The lowest terminal pressure here is 7.5 pounds, and as it is but 8 pounds at the other end, the difference in temperature being but 3°, we will take the lowest as before.

If the difference was greater we should take the average. Substituting for the symbols their values, we have

$$\frac{180+988-100}{100-70}=35.6 \text{ volumes.}$$

The diagrams call for 175.75 gallons per hour, which is one volume, and 35.6 volumes equal 175.75×35.6=6256.7 gallons per hour. When the temperature of the steam is given, instead of the pressure, the formula is as follows:

$$\frac{l+T-t}{t-w}=V.$$

In which l=latent heat, T=temperature of steam, t=temperature of discharge, w=temperature of condensing water, and V=volumes. Substituting for the symbols their values as above, we have

$$\frac{988+180-100}{100-70}=35.6 \text{ volumes.}$$

This operation is practically the same as the one preceding it, but both are given in order to cover all the ground.

The next question that naturally suggests itself in this connection is, how shall we determine the area of the pipe that is to convey this injection water to the condenser? It may be determined by the following formula:

$$\frac{A\ C}{V} = X$$

In which A=volume of water required for generating the steam used, expressed in cubic inches per second, C=number of volumes of water required for condensation, V=velocity due to flow in feet per second and X=area in square inches. This is a very innocent looking formula, but it will require some patience on the part of any one not familiar with it, in order to fully understand the meaning of every part of it. However, it is not very complicated, although the explanation may seem somewhat lengthy.

The first step is to read the formula, in order to avoid the possibility of a misunderstanding here, which we will do as follows: Multiply A by C and divide the product by V, and the quotient will be the area of the pipe in square inches. We will now determine the value of A by use of the following rule: To compute volume of water in a given volume of steam, multiply the volume of steam in cubic feet by its density, and the product will be the volume of water in cubic feet. By referring to Chapter 30 we see that the engine from which the diagrams illustrated there were taken calls for 367.5312 cubic feet of steam per minute, but as the time mentioned in this problem is seconds, we will divide by the number of seconds in a minute, and 367.5312÷60=6.12552 cubic feet per second.

Here a few words concerning the density of steam will not be out of place, for in the tables of properties of saturated steam that are published in the various books of reference, there is one column under the heading of "Density or weight of one cubic foot." This may mislead the casual

reader into thinking that density and weight are synonomous terms, but such is not the case, for the weight refers to the actual weight of a cubic foot of steam, while density means the weight of it as compared with the weight of a cubic foot of water, and taking this as 62.5 pounds, it is a problem in proportion as follows: As 62.5 is to 1, so is the weight of the steam as to its density. In the case in hand, the pressure is 26.5 pounds absolute, as before shown, and we have already demonstrated that a cubic foot at this pressure weighs .06615 pound, therefore 62.5 : 1 : : .06615 : .0010584, which is the density at stated pressure.

As will be noted, the process consists of simply dividing the weight of the steam per cubic foot by 62.5, although this does not explain the reason for the proceeding, as when viewing it as an example in proportion, and it therefore follows that when the density is given and the weight desired, the process is reversed, or in other words, substitute multiplication for division, so that the density multiplied by 62.5 will give the weight.

CHAPTER XXXII.

DETERMINING THE AREA OF INJECTION PIPE.

Proceeding with our problem we have 6.12552 × .0010584 = .006483 cubic feet of water per second. As we wish to have it in cubic inches, we multiply by the number of cubic inches in a cubic foot, and .006483 × 1728 = 11.2 cubic inches per second. Adding 10 per cent. for leakage of valves, etc., makes it 12.32, which is, therefore, the value of A. The value of C is 35.6 as demonstrated in Chapter 31.

In determining the value of V, it is necessary to know the vacuum in the condenser, and in this case it is stated at 25 inches. To reduce to pounds, divide by 2.04, and 25 ÷ 2.04 = 12.25 pounds. To ascertain flow of water in feet per second due to this vacuum, multiply by 2.31 in the case of fresh water, 12.25 × 2.31 = 28.29 feet. To this must be added the vertical distance or height from centre of opening in condenser to surface of water above the condenser, assuming that it is above it.

This is not stated on the diagrams taken from this engine, but we will assume it to be eight feet. Then 28.29 + 8 = 36.29 feet per second. Thus we have established the value of V in the formula, and when we substitute for the symbols their values, it stands as follows:

$$\frac{12.32 \times 35.6}{36.29} = 12.1$$

square inches, or practically a pipe four inches in diameter will be needed for this case, provided the water supply is near at hand, but if a long pipe is necessary, a larger size should be used. The author would call attention to the fact that this formula is a modification of one found in a certain standard book of reference, but it will answer for all ordinary conditions, and for small and medium-sized engines. It may be well to write it

$$\frac{A \quad C}{V \quad D} = X$$

in which case D would be a variable factor, to be changed to suit certain conditions, at the judgment of the engineer.

There is another way to determine the value of A, as follows: The amount accounted for by the indicator is 175.75 gallons per hour. Dividing this by 60 gives us 2.929 gallons per minute, and dividing this quotient by 60 gives us .0488 gallons per second. As there are 231 cubic inches in a gallon, $.0488 \times 231 = 11.2$ cubic inches per second, and adding 10 per cent. makes it 12.32 as before. Here it will be noted that the value of V is calculated on a fresh water basis, but if salt water is to be used, as we may with a surface condenser, then we must multiply the vacuum in pounds by 2.24, instead of 2.31, for, as salt water is heavier, it does not require so high a column to correspond to a stated amount of vacuum.

And again we have assumed that the water of condensation is above the condenser, but it often is below it, which changes the conditions, for in the former case the distance in feet must be added to the height due to the vacuum, as it helps the water in its passage, but in the latter case, with

the supply below the condenser, it must be subtracted from it, as it tends to retard the flow of the water. Suppose then that we assume that we are to use sea water, which is eight feet below the centre of the opening in condenser. In that case the value of V would be $12.25 \times 2.24 - 8 = 19.44$, making the problem stand thus:

$$\frac{12.32 \times 35.6}{19.44} = 22.56$$

square inches, or in other words, requiring a pipe about five inches in diameter.

This conclusion is only a reasonable one, for every engineer knows that when a pump has the water delivered to it under pressure, the pipe may be much smaller than when it must be raised in the ordinary way. From the foregoing calculations it will be seen that it is very necessary to take great care in taking a diagram from an engine, to see that the indicator is in good order and that it is well lubricated. The paper should be smooth and tough, although not necessarily glazed. The pencil should be sharp that it may make a fine line, and so adjusted that it will not bear too hard, for in that case the card or diagram will be distorted, and prove worse than useless.

It is for the best interests of every man now in charge of a steam plant or who expects to be in the future, to understand the use of this instrument, for although the engineer in charge of a small plant may not see the need of such knowledge, as he may be laboring under the impression that it is only necessary for those who are running large engines, yet if the opportunity should present itself for him to be advanced to a more responsible position, provided he understood the indicator, it would not be a proper time for him

to begin to learn then, for while he would be getting posted on the subject, another man would secure the position. Qualify for the position first, and seek the same afterward. Always keep the horse before the cart, and never attempt to reverse the order.

CHAPTER XXXIII.

THREE SYSTEMS OF STEAM HEATING FOR BUILDINGS.

It is well for the applicant for a license to understand something of steam heating systems, and their application to use in every day practice, as this will be necessary in many cases, for this apparatus will be under care of the engineer, and, although he may be able to run a certain system from day to day, provided nothing unusual happens, still, it is much better for him to understand the principle on which it works, for then he can more readily detect the cause of failure of any part of it to work. The writer well remembers presenting the question: "What is the meaning of the term indirect radiation?" before an assembly of engineers, and not one of them could give a satisfactory reply, therefore it may be well to give a description of the three principal systems of heating buildings, namely, direct radiation, indirect radiation and direct-indirect radiation.

The first is just what its name applies, and that is that the heat is radiated directly into the room to be warmed from coils or radiators located in said room. The pressure may be high or low, and steam or hot water may be used, but so long as the radiators are in the room to be heated, and are not connected with any system of ventilation, this system is known as direct radiation.

The indirect radiation system is one in which the air is warmed by coils or banks of pipes located in some place outside of the room to be heated, and then forced by means of a fan into rooms where it is needed, or small separate radiators may be placed near the bottom of the flues leading into the rooms, when, as the air becomes heated, it naturally rises, and thus can be utilized.

It is quite evident that the fan is to be preferred, as it makes the circulation positive, and does not depend on circumstances, such as "which way the wind blows," etc. From the nature of the system it will be noted that it can only be used in connection with some system of ventilation.

The direct-indirect system, as its name indicates, is something of a combination of the other two systems, for it is one in which the radiators are partly or wholly within the room to be heated, and are in direct connection with some system of ventilation.

The indirect system is used at the present time for heating factories more than ever before, and the apparatus is very simply, consisting of a large coil or bank of pipe through which the exhaust steam from the engine circulates, or when it is not running, live steam from the boiler is introduced. The coil is encased in a jacket of sheet iron, or of something more substantial if it is very large, and a fan blower forces the cold air into this jacket, where it is heated, and is then led away by means of suitable pipes to the different rooms where it is needed. The fan is often run by a small independent engine, and so equipped that the speed may be varied at will. One great advantage of this system is that while it heats it also ventilates, and in summer time cool air may be forced through, and the rooms rendered more

comfortable. Care should be taken to have the inlet for air in such a place as to secure that which is pure and wholesome, as the odor of the rooms will be unpleasant otherwise.

The direct-indirect system reminds one of a factory, the main part of which has been outgrown by the business carried on therein, and to provide necessary room, additions are built on from time to time, until the whole affair has a patched-up appearance. So it is with this system, for while there is no doubt about its giving good results in many cases, still it is not as symmetrical as the others, and involves unnecessary complications.

The direct radiation system for steam is probably used more than any other for heating public buildings, and as there is more than one way to pipe up this system, we would call attention to this fact and to some of the methods adopted. We frequently hear the question asked: "Can the water of condensation from coils and radiators be returned to the boiler without the use of some mechanical contrivance made for this purpose?" To this we would reply that it can and is done in many places, and we see no objection to it.

Of course the radiators must be above the water line of the boiler, and it is a good plan to have the piping so arranged that the cold water which fills the returns when steam is first turned on may be blown out into the sewer (through suitable connections), and when the body of water is in motion and circulation established, the blow-off valve may be closed, when the water will be returned to the boiler. The author is aware that this has been tried, and in some cases proved a failure, but wherever it has there is some straight reason for it. It is necessary to maintain boiler pressure in

the radiators, and if there are so many of them that the opening in the dome or the intermediate pipes are not large enough to supply sufficient steam to do this, then the water of condensation cannot return to the boiler, unless it is pumped in. This is the principal cause of failure.

If some of the radiators must be located lower than the water line, it will be necessary to provide a pump or trap to return the water of condensation.

CHAPTER XXXIV.

COMPARISONS OF SYSTEMS FOR STEAM HEATING.

The system of direct radiation for steam heating may be piped up in four different ways as follows: (1) A main steam pipe is continued from the dome of boiler up to the highest floor of the building to be heated. From this pipe, which is commonly called a riser, steam is taken for radiators located on each floor, and separate return pipes are provided for each, which are not connected together until fhey are brought down below the hydrostatic level. (2) A riser is put up and connected in the same way as before, but the drip pipes are all connected into one pipe, called the return riser, in the same way that steam is taken from the steam riser, and thus returned to the boiler. (3) The steam riser is provided as before, which is also made to do duty as a return riser until it reaches the point where it is connected into the horizontal steam pipe from boiler. Here a tee is used instead of an elbow, and the return pipe is continued down until it is below the hydrostatic level, when it is carried in a horizontal direction to the boiler. (4) A single pipe is the only connection between boiler and radiator, in which the steam passes along the upper part of it, and the water of condensation returns along the lower part of it. As a matter of course all of the pipes in this circulation must incline towards the boiler, or else it will not work at all.

System No. 1 is to be recommended above the others, for it gives practically a positive circulation, and especially where low steam pressure is to be used, and in places where it will be run by persons who are not experts, as there should be no trouble in expelling the air, and preventing all of the disagreeable cracking noises frequently heard in other systems, caused by steam coming in contact with water, etc. Its first cost is a little more than either of the others.

No. 2 is used much in large buildings, and gives fair results when properly put up and cared for. It requires less labor and material, consequently the expense is less.

In No. 3 much larger pipes must be used, and for that reason the cost in some cases will equal that of No. 2, and the results will be less satisfactory. The same objections apply to No. 4, and therefore it is not to be recommended except for small jobs, or where for some special reason it may be desirable.

Where Nos. 1 or 2 are used care must be taken to provide a drip connection, so that no water can collect in the vertical steam supply pipe or steam riser, for it must be remembered that the flow of steam through this pipe is often very slow, and consequently the condensation is excessive.

It is not intended to give a complete description of the details of piping for these systems, as it would require a volume of reading matter and many illustrations, but the few hints and suggestions here offered will be found of practical use, as for instance it is better to use angle valves wherever practicable, as they give full opening, cause less friction and save the use of an elbow,

Where a valve must be located in a straight pipe, a straight-way valve is preferable, but if for

any reason a globe valve must be used, then do not place the stem in a vertical position, for if it is, the water running along the bottom of the pipe will be trapped there, owing to the construction of the valve, and much unnecessary strain will be brought to bear on the pipe by the hammering action of this water.

Pieces of pipe that have been broken by this action, under an ordinary working steam pressure, have afterward been tested, and withstood 1000 pounds pressure applied in a careful and steady manner.

When steam is first admitted to a system of piping, the valve should be opened very slowly, as the first steam that enters will be condensed, causing what is called "water hammer," therefore it should be admitted as carefully as possible. If a globe or angle valve is used, it is much easier to tell just when it begins to open than it is in the case of a gate valve, and, as it frequently happens that we only wish to admit a little steam to a radiator in moderate weather, this is one thing in their favor. This refers to radiators having independent drips.

The engineer who is systematic and careful about his work, and takes pride in seeing it properly done, always likes to have a valve stem placed in either a vertical or horizontal position, and with a globe valve, the latter is the best of the two, but neither one of them are just right so far as service is concerned, for if the former plan is adopted, the water will be trapped in the pipe, owing to the construction of the valve, and if the latter is, the water will stand in the bonnet and leak out around the stem, therefore if the stem is inclined upward slightly it will give the best results, but at the same time does not look as well.

Valves should be put on with the bottom towards the source of pressure, so that when they are shut there will be no steam on the stuffing box, for otherwise the valve cannot be packed without relieving the whole system of pressure. Asbestos wicking for packing valve stems is far preferable to the ordinary candle wicking often used for that purpose.

CHAPTER XXXV.

DETERMINING THE RADIATING SURFACE NECESSARY TO HEAT A ROOM OR BUILDING.

If the candidate for a license is called upon to give a rule for determining the heating or radiating surface necessary to warm a certain room, he should reply that there can be no inflexible rule given for such a purpose, as it will depend largely upon the conditions existing in each individual case, such as the location of the room, the manner in which the walls are built, the number and size of windows that it contains, and the temperature that must be maintained in it. There are, however, rules that may be used for the foundation of such a calculation. The following is said to be in use among some of the steam fitters who put in heating apparatus, and many consider that it gives good results.

Multiply the length, breadth and height of the room, in feet, together, and from the product so obtained cut off two figures from the right hand, and call the remainder square feet of heating surface, or, as some might call it, radiating surface. This is for a high pressure system, but for a low pressure one additions must be made according to the judgment of the engineer in charge, for an exposed room will need more heat than one that is well protected, and if there are many windows in it, that will also be a consideration, for glass is a good conductor of heat. For an illustration I

shall take a room in the immediate vicinity of where this is written.

It is 30.5 feet long, 17 feet wide and 11 feet high; $30.5 \times 17 \times 11 = 5703$ cubic feet. As about two-thirds of this room is exposed, we will add 50 per cent. on this account, and we have $5703 + 2851 = 8554$. Cutting off the two right hand figures leaves 85 square feet of heating surface, or about 247 feet of one inch pipe. This rule, however, is not a very satisfactory one, as it does not take into account all existing conditions. It may be called a "rule of thumb."

Now while a glazed window admits the heat of the sun, if it is shining directly on it, more than a brick wall or other opaque surface will, still, if the sun is not shining on it, then the glass becomes an important factor in the calculation, as it will cool off the room much faster than brick walls, sheathing or lath and plaster will. The following rule has the endorsement of an eminent authority on steam heating: Divide the difference in temperature between that at which the room is to be kept and the coldest outside atmosphere, by the difference between the temperature of the steam pipes and that at which you wish to keep the room, and the quotient will be the square feet, or fraction thereof, of plate or pipe surface to each square foot of glass or its equivalent in wall surface.

We will now explain the way to reduce wall surface to its equivalent in glass surface, and for this purpose we will introduce the following table from "Steam Heating for Buildings," by William J. Baldwin. Table of approximate power for transmitting heat, of various building substance compared with each other:

Window glass	1000		
Oak and walnut sheathing on walls	66	to	100
White pine and pitch pine	80	to	100
Lath and plaster walls, good	75	to	100
Lath and plaster walls, common	100	to	105
Common brick (rough)	150		
Common brick (hard finish)	200		
Common brick (hard finish, hollow walls)	150		
Sheet iron	1100	to	1200

"In figuring wall surface, etc., multiply the superficial area of the wall in square feet by the number opposite the substance in the table and divide by 1000 (the value of glass). The product is the equivalent of so many square feet of glass in cooling power, and may be added to the window surface and treated in the same way."

Let us now apply this rule to the case of the room mentioned before and note the result. It is 17×30.5 feet, so that there are 518.5 square feet in the ceiling, and the same amount in the floor, making 1037 square feet in both. This surface is taken into account because the room is in an isolated building comparatively, there being no other rooms above or below it to assist in keeping it warm.

In the sides of this room there are 30.5×11×2 =671 feet. In the ends 17×11×2=374 feet, making 2082 square feet in all. A part of this is glass surface and the remainder is hard pine sheathing, and as the two represent different values, we must separate them and reduce them to a common standard. There are seven windows, each containing 11 square feet of glass, and one sash door with nine feet. 11×7+9=86 square feet; 2082−86=1996. Thus we have 86 square feet of glass, and 1996 square feet of hard pine sheathing to provide for.

The latter we will reduce to its equivalent in glass surface, and add the two together. As this table tells us that the pitch pine sheathing is valued at from 80 to 100 we will take it at 90 for an average. $1996 \times 90 \div 1000 = 179$, which is the surface of the sheathing in the room reduced to its equivalent in glass surface. To this must be added 86 feet of glass surface, and $179 + 86 = 265$ square feet.

CHAPTER XXXVI.

AREA OF MAIN STEAM PIPE FOR DIRECT RADIATING SYSTEM OF STEAM HEATING.—SIZE OF BOILER REQUIRED.

We will now proceed to assume conditions in order to enable us to finish working out the example presented in Chapter 35. Assuming that the coldest outside temperature is at zero, or at 0°, and we wish to warm the room so that the thermometer will stand at 70°, we have 70°−0= 70°. And here the question presents itself as to what we shall do if the coldest outside temperature were taken at 10° below zero. In that case we would subtract the difference, as before, and add 10 to the remainder. A practical way to decide it is to take a thermometer in hand and calculate or count the difference in degrees. If we carry 60 pounds pressure on our heating pipes, the temperature will be 307°.

The temperature of the room being 70, we have 307−70=237, and 70÷237=.295. Add 50 per cent. for ventilation, air leaks, etc., and .295+.147= .442, which is the number of square feet of heating suface needed for each square foot of glass surface, or its equivalent. In this case it is 265, and 265×.442=117 square feet, or about 340 feet of one-inch pipe. Now let us assume different conditions and compare results. Suppose that the difference between the inside and outside temperatures is 70°, and we are using steam at two

pounds pressure. Then we would have 219−70=149, and 70÷149=.47. Add 50 per cent. as before, and it brings it up to .705 square feet of radiating surface for each square foot of glass surface, or its equivalent, 265×.705=186 square feet, or about 540 feet of one-inch pipe.

Now suppose that we wish to estimate what piping, boiler capacity, etc., will be needed to heat a building containing 12 rooms like the one before mentioned, the system to be direct radiation, and low pressure, say two pounds. In such a case we would not figure in the floors and ceiling, as they would not be exposed as in the case of an isolated room; 2082−1037=1045 square feet in the sides and ends of one room. In 12 rooms there would be 1045×12=12,540 square feet. In each room there would be 86 square feet of glass, and 86×12=1032 square feet; 12,540−1032×11,508 square feet of pine sheathing to be reduced to its equivalent in square feet of glass, and 11,508×90÷1000=1035; 1032+1035=2067 square feet of glass, or its equivalent.

Assuming a difference of outside and inside temperatures amounting to 70° as before, and a difference between the pipe and the room of 149°, and our multiplier is found to be .47, or if we wish to add the 50 per cent. here it makes it .705. Now 2067×.705=1457 square feet of radiating surface needed, which would be about 4229 feet of one-inch pipe.

Our next move is to determine the area of the main steam pipe, which, of course, must be based on the number of square feet of radiating surface that it must supply steam for, and to the number already obtained must be added the surface contained in the main itself.

This we can only assume, but if we call it 20 per

cent. of the other, it will answer our purpose. This amounts to 291 square feet, and 1457+291=1748 square feet. Competent authorities tell us, and practice shows them to be correct, that for each 100 square feet of heating surface in the building, the main steam pipe should contain the area of a one-inch pipe, or .7854 of a square inch. 1748÷100=17.48×.7854=13.7 square inches. Now the actual internal area of what we ordinarily call a four-inch pipe is stated at 12.7 square inches. Therefore a four-inch pipe will answer our purpose.

Another way to tell what the diameter of the main steam pipe should be, is as follows: Having found the number of square feet of heating surface needed, extract the square root of it, and one-tenth of this number will be the desired diameter. The square root of 1748 is 41.8+ and one-tenth of this is 4.18, or a four-inch pipe as before. Probably many buildings of this size are supplied with steam by a smaller pipe than this, but it does not necessarily follow that the best results are obtained thereby.

In regard to the amount of heating surface in the boiler that will be needed to supply steam for these rooms, if there is one square foot of surface exposed to the direct action of the fire, for each six square feet of radiating surface, it will be sufficient for the conditions before mentioned, provided the firing is attended to in a proper manner. This would call for a boiler containing about 300 square feet of direct heating surface. If a portion of it is tube or flue surface, a large allowance must be made, as its efficiency is much less. If a horizontal multitubular boiler is to be used, it should be four feet in diameter and 12 feet long, having, say 36 three-inch tubes. This

gives 150 feet of direct heating surface, and about 300 feet of tube surface.

It should be noted that the data given here is for a system of direct radiation for steam.

CHAPTER XXXVII.

HEATING WITH EXHAUST STEAM.

Occasionally we meet an engineer who does not appreciate the value of exhaust steam for heating purposes, as they allow it to go to waste, or at least they only utilize a portion of it in heating the feed water, and although it does not pay to cause excessive back pressure on the engine, still if the system of piping is properly arranged, all of the exhaust steam left after passing through the heater, may be used during the winter season with very little detriment on account of back pressure. The cost of such steam in any case may be calculated in the following way:

Ascertain the horse power constant of the engine, and multiply it by the back pressure due to the weighting of the back pressure valve. The result will be the horse power expended in forcing the exhaust steam around the factory. Suppose that we have an engine 24x60 inches running at 60 revolutions per minute. When the back pressure valve is up or open, the back pressure is one pound above the atmosphere, and when it is down or closed, it is two pounds above it, what does it cost to use the exhaust steam for heating? The horse power constant for this engine is 8.225, so that when we multiply it by the one pound additional back pressure caused by closing the back pressure valve, and forcing the steam through the heating pipes, we see that we can utilize a large volume of steam at a very slight cost.

If the piping is properly arranged this can be accomplished, but at the same time we do not recommend that any engineer figure it up in his own case on this basis, unless he knows just what the back pressure is, for it is quite possible to have matters so arranged that it will cause 10 pounds back pressure instead of one.

In many cases the exhaust pipe is carried up to the top of the factory, even when it is a high one, in order to get the exhaust steam away from the windows, and if it is provided with ample drips, or rather drip pipes, located at its lowest point, the steam may be carried up and cause very little back pressure, as it does not take much to force steam up, but, on the other hand, it requires much more to raise water. Cases can be found where the drip pipe is located at some point convenient of access, without regard to whether it is at the lowest point or not.

While such an arrangement may not cause any part of the exhaust pipe to entirely fill with water, still if there are two or three inches of water standing in an eight-inch pipe it will cause the steam at the end of the exhaust pipe to appear very wet, giving rise to the idea that the boiler primes badly, when such is not the case, and causing unnecessary back pressure on the engine.

With the exhaust pipe carried up at one end of the building, and tees having been put in at suitable places, say eight feet above each floor, if the ceiling is high enough to admit of it, then nipples may be put in, and other tees connected, from which branch pipes may be carried to either side of the room, where they are connected into a coil on each side. This is preferable to one large coil over the centre, not only on account of a better distribution of the heat, but because by placing

valves at the inlet and outlet of each coil, one of them may be shut off in moderate weather, and where the building is so situated that the sun shines on one side of it during the forenoon, and on the other side during the afternoon, the steam may be put on the coldest side of room, thus nearly equalizing the temperature of the same. These coils should all gradually incline from the main exhaust pipe toward the other end of the rooms, allowing them a pitch of about one-half inch for every 10 feet in length. At the other end of the factory there should be a vertical pipe of about one-quarter of the diameter of the exhaust pipe, into which all the drips from the coils should be connected. The lower end of this pipe should be attached to a good steam trap, for then no steam will be blown out near the windows. In this pipe, above the highest drip connection, there should be a valve, and from the valve the pipe should be continued up through the roof. Thus the air will be readily removed when the engine is started and the back pressure valve is closed, and the trap will take care of all of the water. If it is desired to put on live steam when the engine is not running, the valves near the exhaust pipe may be closed, and also the highest valve in the vertical drip pipe, or what might be called the main return riser, and the live steam turned on at pleasure.

CHAPTER XXXVIII.

DIAMETER AND HORSE POWER OF SHAFTING.

A knowledge of the strength of shafting of various kinds may be required in this connection, and it is well to remember that a shaft of wrought iron or any other material will transmit a given horse power under certain conditions and last for years, while the same shaft would soon fail under other conditions, although the power transmitted is the same in both cases, or even less in the latter.

The following formula is given us for determining the diameter of a wrought-iron shaft to transmit a given horse power under favorable conditions, namely, as in the case of a main shaft in a mill or factory, where it is well supported, not subjected to excessive strain from belts being tighter than they should be and kept in line. It is one that agrees as well with good practice as any that can be found:

$$\sqrt[3]{\frac{50 \text{ I. H. P.}}{n}} = d$$

in which I. H. P. is the indicated horse power to be transmitted, n = number of revolutions, and d = diameter of shaft. It reads as follows: Multiply 50 by the indicated horse power, and divide the product by the number of revolutions per minute. The cube root of the quotient is the diameter of the shaft in inches.

For example let us take the case of a shaft made

of wrought iron which we wish to have tramsmit 138 horse power, running at 115 turns per minute. Substituting for the symbols their values in this case we have:

$$\sqrt[3]{\frac{50\times138}{115}} =$$

3.914 inches, or as the sizes ordinarily run, we would call it a 3 15-16 inches shaft. If the size of shaft and number of revolutions are given, and the power that it will transmit is required, the formula is as follows: $.02\ n\ d^3 = \text{I. H. P.}$, which reads, multiply .02 by the number of revolutions per minute, and the product by the cube of the diameter.

The product so obtained will be the horse power that the shaft will transmit.

Suppose that we have a 3½ inch wrought-iron shaft whose speed is 100 revolutions per minute, and we wish to know how much power it will safely transmit: $.02\times100\times3.5\times3.5\times3.5 = 85.75$ indicated horse power. It will be noted that the power increases directly as the speed is increased, for if this shaft should be run at 200 revolutions, it would easily transmit 171.50 horse power. If the shaft is made of steel instead of wrought iron the formula is as follows, the letters denoting the same as in the preceding:

$$\sqrt[3]{\frac{31.25\ \text{I. H. P.}}{n}} = d$$

Assuming the elements of the preceding case, except that the shaft is made of steel, and substituting for the symbols their values, we have:

$$\sqrt[3]{\frac{31.25\times138}{115}} =$$

3.4—inches nearly. If the size of shaft and speed is given. and the power that it will transmit is required in the case of a steel shaft, the formula is as follows: $.032\ n\ d^3 = I.\ H.\ P.$ With a shaft running at 100 revolutions per minute, whose diameter is 3.5 inches we would have $.032 \times 100 \times 3.5 \times 3.5 \times 3.5 = 137.2$. (The author's experience with steel shafts has been that they are apt to prove treacherous as they sometimes break under very light loads, on account of hidden flaws).

If we wish to use a cast iron shaft, the formula for determining the diameter of it is as follows:

$$\sqrt[3]{\frac{83.5\ I.\ H.\ P.}{n}} = d$$

the letters denoting the same as above. Assuming the elements of a preceding case, and substituting for the symbols their values gives us the following:

$$\sqrt[3]{\frac{83.5 \times 138}{115}} =$$

4.641 inches, which is the diameter of the cast iron shaft. If the number of revolutions per minute, and the diameter is given, and the power that it will transmit required, the formula is as follows: $.012\ n\ d^3 = I.\ H.\ P.$ Assuming the elements of a preceding case and we have $.012 \times 100 \times 3.5 \times 3.5 \times 3.5 = 51.45$ indicated horse power.

The foregoing rules and formula do not apply to crank shafts of steam engines, propeller shafts, or other conditions where the service required is severe. For determining the size of crank shaft for compound engines, with cranks set at an angle of 90° we have the following formula for wrought iron:

$$\sqrt[3]{\frac{D^2p+d^2 15}{2468}\times S} =$$

diameter of shaft, in which D is the diameter of high pressure cylinder, in inches, p is the boiler or steam chest pressure above a vacuum, d is the diameter of low pressure cylinder in inches, and S is the stroke in inches. This may be read as follows: Square the diameter of the high pressure cylinder, and multiply by the steam-chest pressure absolute. To this product add the square of the diameter of the low pressure cylinder multiplied by 15 and divide their sum by 2468. Multiply the quotient by the stroke in inches, and extract the cube root of the product which will be the diameter of the shaft in inches. For the sake of illustrating this formula we will suppose that a cross compound engine with cylinders 20 and 40 inches in diameter, respectively, and a stroke of 48 inches to be operated under a pressure of 140 pounds absolute, is to be used, and we wish to know what the diameter of the crank shaft should be. Substitute for the symbols their values and we have the following:

$$\sqrt[3]{\frac{20^2\times 140+40^2\times 15}{2468}\times 48} = 11.6 \text{ inches nearly.}$$

In practice a 12 inch shaft would be used.

As we wish to make every part of this book plain as we proceed, we will give the solution of this problem in detail. The diameter of the high pressure cylinder is 20 inches, which being squared gives us $20\times 20=400$. The absolute pressure is 140 pounds, and $400\times 140=56{,}000$. The diameter of the low pressure cylinder is 40 inches, which

being squared gives us 40×40=1600, and multiplying by 15 makes it 24,000. This is to be added to our first product, and 56,000+24,000=80,000, and 80,000÷2468=32.41. This is to be multiplied by the stroke in inches, and 32.41×48=1555.68, the cube root of which is 11.6 nearly.

CHAPTER XXXIX.

WIDTH, SPEED AND HORSE POWER OF BELTING.

Probably there are few subjects connected with steam engineering concerning which there is a greater diversity of opinion than as to a proper rule for determining the horse power that a belt will transmit, and also for giving the proper width of belt to transmit a given amount of power. The writer does not expect to settle the question by anything that may be written here, but at the same time does not hesitate to give a formula that takes into consideration all, or nearly all of the factors in the case. It is taken from Nystrom's Mechanics and is to determine the power that a belt will transmit, and is as follows:

$$\frac{B\,d\,n\,Z\,S}{15{,}000{,}000} =$$

= H. P., in which B = breadth of belt in inches; d equals diameter of the smallest pulley in inches; n = revolutions per minute of the smallest pulley; Z = half angle of contact of belt on smallest pulley; S = safe working strength of belt in pounds per inch of width, for ordinary single leather belt is taken at 100 pounds, and H. P. = the horse power that can be easily transmitted. For example, we will take the case of a belt 48 inches wide, running over pulleys of which the smaller one is 54 inches in diameter, and revolves 200 times per minute. Half angle of contact is 85

degrees, and safe load 100 pounds per inch of width. Our problem then becomes:

$$\frac{48 \times 54 \times 200 \times 85 \times 100}{15{,}000{,}000} = 293.7$$

horse power. The reason for taking the width of belt into account is because each inch or fraction thereof will transmit a certain amount of power. The diameter of smaller pulley is taken because that effects the friction of the belt. The speed of it is noted because that effects the power transmitted directly, just the same as the speed of shafting effects it. The term "half angle of contact" may need some explanation. The rim of the pulley is a circle, and accordingly contains 360 degrees. If both of the pulleys were the same size, and there was no sag to the belt, it would lap around one-half of this circle, or 180 degrees, but as the other pulley is assumed to be larger than this one, and the belt sags a trifle (which is a help in some cases, and a detriment in others), the angle of contact is taken at 170 degrees, and one-half of this is 85 degrees. The safe load is determined by experiment, but for belts in good order may be taken at 100. The divisor is a constant number.

If we wish to determine the breadth of belt necessary to transmit a given horse power, our formula becomes

$$\frac{15{,}000{,}000 \text{ H. P.}}{d\, n\, Z\, S} =$$

breadth, the letters meaning the same as before. Assuming elements of the preceding case, we have

$$\frac{15{,}000{,}000 \times 293.7}{54 \times 200 \times 85 \times 100} = 48 \text{ inches.}$$

These formulæ apply to leather belts in good order, running on clean iron pulleys. There are several possible conditions which tend to modify them, as, for instance, if a double belt is running over wooden pulleys, or iron pulleys covered with leather, the efficiency will be greater, and if the belts are hard and dry, and running over pulleys that are very rough, or extraordinarily smooth on their faces, the efficiency will be decreased.

If we have a good double belt instead of a single one, the value of S will be doubled, or taken at 200 pounds, and accordingly the horse power that it will transmit will be doubled and $293.7 \times 2 = 587.4$ horse power.

This assumes that the belt will not slip on the pulley even if the load is increased until the full safe working strain on the belt is reached. If a belt is cemented and riveted together, or in other words is made "endless," it will be much stronger than if it is laced in the usual way, for reasons that are plain to every engineer who has given the matter any consideration.

CHAPTER XL.

STEAM BOILER EXPLOSIONS.

The applicant should be well posted as to the causes, or perhaps we might be allowed to say the cause of steam boiler explosions, and also as to the theories which have been advanced at one time or another, for the same, but which can hardly be called correct ones in view of other facts in the same connection. Take for instance the theory of superheated water as an active agent in this matter.

The advocates of this theory claim that globules of water absorb the heat of the fire in the furnace, and that while in a state of rest they are harmless, but when they rise to the surface and burst from some unknown cause they develop a vast amount of energy, so much in fact, that no boiler can withstand the accumulation of it.

This is said to be the condition of things when we shut down our engines, for while they are at rest it is assumed that the heat is being generated in the furnace, very much the same as when the machinery is in motion, and as the engine is not taking it, it is silently stored up for future use or abuse. So far as our observation and practice have been teachers, they have taught us that when a fire is properly banked previous to shutting down the engine there is very little heat generated to go anywhere. We are well aware that in places where it is necessary to force fires almost up to

shutting-down time the brickwork, arch plate, dead plate and other parts not in contact with water are at a high temperature, and some steam will be generated by this heat after the fires are banked, but even then the only effect that we will have noticed is that the lever safety valve slowly lifts as the pressure accumulates, and the surplus passes out into the atmosphere, and we cannot see how it can do any more harm to the boiler than if the steam was used by the engine. But we are told that when the throttle valve is opened to start the engine the pressure on the top of the water is relieved, and it therefore rises and is converted into steam of great force, so that no gage can indicate it.

We wonder how it is that anyone can still advocate this theory after they have stood by the side of one of our first class passenger locomotives, as they stand at a station ready to start out on a swift journey, with its train of half a dozen coaches, for while it waits the heat is being rapidly absorbed by the water, and the pressure rises until the safe working pressure is reached, when the pop safety valve opens with such celerity as to cause the unsuspecting casual observer to give an involuntary jump, but no explosion takes place, and in addition to all this, certain kinds of engines are started up several times each day, and suddenly, too, but no explosion follows. Cases can be cited, it is true, where explosions have taken place just after the engine was started up, but what does that signify? Boilers appear to burst just as easily and as frequently, too, when the engine for which they furnish steam have been running an hour or two, as they do when they are just started, and sometimes they explode during the night when no steam is being used, and others ex-

plode when they never were used for supplying an engine.

We are told that electricity accumulates in a boiler until it finally gets so full of it that an explosion follows, but it appears as if in order to hold a large volume of electricity, a boiler would of necessity have to be set so that it would be insulated from all surrounding objects and we never saw a boiler set in that way to be used for ordinary purposes. We suppose that if a boiler were carrying a heavy pressure and should be struck by lightning that the concussion together with the heavy pressure might cause an explosion, but that is a different thing. The favorite cause ascribed is that there was not sufficient water in the boiler to prevent the catastrophe, for it is claimed that the water having got low, the plates became red hot, and when cold water was pumped in, it immediately flashed into steam, which accumulated faster than the safety valve could dispose of it, hence the trouble.

The first point that we wish to call attention to in this connection is this: "Do the plates of an ordinary boiler contain heat enough to cause a large body of water to evaporate so rapidly?" Is this not worthy of consideration? The second point is that in the case of an ordinary tubular boiler, the water would have to be very low before the plates could get red hot, and if the feed enters through the bottom of the shell the boiler would have to be entirely empty before cold water could be pumped on to the red hot sheets? We do not advance the claim nor make the assertion that insufficiency of water cannot cause a boiler to explode, for we believe that it will under certain conditions.

If the iron in a boiler becomes red hot, it is not

so strong as it is when as hot as normal conditions, while in use make it, and then it might give out under an ordinary working pressure, but an examination of the wreck would surely disclose the cause of the disaster.

Experiments have been made to show the truth or falsity of the red hot sheet theory, and in every instance where the cold water was pumped on to the red hot plates, a very natural result followed, and that is that the plates cooled off. The rise in pressure was very slight, or else conspicuous on account of its absence in every instance.

CHAPTER XLI.

STEAM BOILER EXPLOSIONS CONTINUED.

If the plates are allowed to get red hot while the boiler is under a full working pressure, and they give out on this account, there would be an elongation of the iron, or perhaps we ought to call it a reduction of the cross section at the point of rupture, which would prove, in connection with the general appearance of the iron, that the plates were red hot if such were the case. However, the sheets in a boiler may be ruined by low water, without causing them to get red hot, for if they are not covered for a short time at once, and the practice is persisted in, the strength of the iron will be lessened, and in time it may become unable to sustain an ordinary working pressure.

Many persons appear to think that low water is the one and only cause of the explosion of steam boilers, and of these we wish to ask a question. Suppose that you have a chain that is capable of sustaining a weight of 4000 pounds and no more. If you attempt to lift a weight of 6000 pounds with it, as a matter of course one or more of the links in it are broken, and the weight is not lifted. Now if you had so arranged matters that the chain was covered with water when the strain was put upon it would it have made it any stronger than when no water was near it? Undoubtedly you will reply in the negative, but did you ever

reflect that if more strain is put upon the iron in a boiler than it can stand that it will break in two, just the same as the iron in the chain did when it was taxed beyond its strength? If a boiler is safe at 80 pounds pressure, and you attempt to carry 150 pounds on it, do not be surprised if the result is disastrous, notwithstanding your boiler is full of water, even to its highest part.

Some people appear to think that if a boiler had three gages of water in it when it exploded, the water would all be there afterwards, forgetting that when the pressure is so suddenly relieved, a portion of it will flash into steam and float away. On the other hand, those who advocate the idea that all of the water will always flash into steam under such conditions are not wholly right, for it cannot be converted into steam unless there is heat enough stored in it to cause it to evaporate, and the necessary heat is not always present. Some claim that if a boiler does explode when there are three gages of water in it, the result will not be disastrous, for as soon as the pressure is relieved, the water will simply run out, and that is all that there is to it, but while some weak boilers may have given out under a low pressure, and the result been as above mentioned, still there was some special cause for it, and it is the exception rather than the rule.

There are several other theories which have been advocated at various times in the past, with a greater or less degree of enthusiasm, but which have failed to secure recognition by theoretical and practical steam engineers, and consequently have been laid on the shelf.

The real cause for all boiler explosions is simply that the structure was not strong enough to stand the strain put upon it.

Of course we shall proceed to specify the different reasons for this state of things, and call attention to some of the practices of boiler makers and boiler users, that work together in perfect harmony to bring about these catastrophes which some people think originate in mystery and end in uncertainty.

We saw an article in an alleged funny paper some time ago, which deprecated the practice of telling what caused a boiler to explode, after it had gone up in a cloud of dust, and expressed a wish that hereafter we might be informed of the dangerous condition of affairs just before the departure of the source of power, so that we should have time to get out of the way. A good suggestion, truly, but while it is not possible to tell just when the last straw will be laid on the camel's back, still we can point out things that should be avoided because they eventually lead to trouble.

There is great rivalry among boiler makers at the present time, as to who can furnish a certain plant for the least money, or build a boiler that will evaporate more water than any other extant, and sometimes it appears as if they were endeavoring to outdo all others in presenting unique designs.

Some of these are not so planned as to be durable in practice, although they may show a high duty when tested for economy. It is not intended to specify any particular kind or style, but some of them are so designed that unequal contraction and expansion cause more strain on the several parts than the steam pressure does, which shortens the life of the fabric, and is a defect which is apt to cause trouble before it is discovered in practice.

The design may be such as to necessitate the

use of large, flat surfaces, which are always to be avoided as much as possible, as they are weak points and must be strengthened with braces and stays, unless said parts are made of cast iron, and then it is almost impossible to guard against blow holes and excessive strains caused by the action of the metal itself in cooling after it is cast. If it is necessary to cut large holes in the shell, as it is with some kinds, it is a source of weakness even after the edges have been reinforced by riveting cast-iron frames on them.

CHAPTER XLII.

STEAM BOILER EXPLOSIONS CONCLUDED.

In some other kinds, series of holes are bored in the shell, into which tubes of some kind or description are expanded, which, without doubt, greatly weakens the shell. In others great dependence is placed on bolts, which are expected to hold good sized parts together against the direct pressure of steam, which is also a weak point. Still others are so made that parts above the water line are exposed to the action of the products of combustion on their way to the chimney, and although it is expected that those products will be comparatively cool by the time that they reach these exposed parts, still they are sometimes hot enough to do much harm.

Poorly constructed boilers are the cause of many explosions, and under this head may be mentioned defective riveting, which is probably the most difficult fault for the running engineer to discover, for boilers have been run for years, and passed several inspections by agents of the insurance company, by whom they are pronounced all right, and afterward, through some mishap, or in some accidental way, it has been discovered that several rivets in a single seam were defective, and undoubtedly had been so from the time that the boiler was built. If the sheets are punched or drilled separately it sometimes happens that the holes do not come fair when they are put together, and to remedy the trouble the drift pin is

inserted and made to do regular duty, the result being that the fibres of the iron are disturbed, and sometimes the plate is cracked. In either of the above cases the result is that there is a weak spot there, and when the wreck is examined after the explosion we are told just where the initial rupture was.

In other cases not enough braces are put in to make the boiler safe under the pressure that it is designed to carry, or they may be improperly located, so that some of them have more strain on them than they can safely carry, or one or more of them might be defective and break, thus causing an excessive load to be put upon the others, causing them to give out also. Sometimes the braces on a flat surface may be in good order and properly located at that end of them, but at the other end they may be bunched together, which concentrates the strain in one place more than it ought to, and trouble follows. Disastrous explosions have been caused by this defective arrangement of these important parts.

Sometimes braces are made in such a way that when a heavy load is put upon them they will straighten out, thus increasing their length, allowing the head to bulge out before they really perform the work that they were intended for.

Some defects in the iron or steel of which the plates are made are so well covered up as to escape detection until they are put into use, but even if a boiler is well designed and properly constructed after it has been used for a time it will show signs of wear. Some steam users evidently are not aware of this fact, for they seem to be of the opinion that because a boiler is made of iron or steel it can never wear out; but when we consider the work that they have to do, and the influences

that are at work to cause their destruction, even when well managed and cared for, it is not surprising that they do not last forever. Take for illustration the case of an ordinary tubular boiler furnishing steam for an automatic, non-cendensing engine, developing 100 horse power. Every hour there are 3000 pounds of water pumped into this boiler, or 15 tons every ten hours. Enough heat must be generated under it to evaporate this large amount of water, and of course it must pass through the iron before it can reach the water. If the plates were always clean it would not be so bad, but when a hard scale has accumulated on them it requires more heat to do the work, as scale is a non-conductor of heat, and consequently the life of the iron is burned out sooner. The fact that this water circulates rapidly must not be lost sight of, for that portion which is directly over the fire rises as it becomes heated, and must be replaced by that from cooler parts, and this is kept up as long as the boiler is in use.

It is an old saying that the "gentle waves wear the solid rock," and will they not wear on iron just as well? In many cases the scale acts as a protector, and so is of some value, but nevertheless we notice that the inspectors report some cases of internal grooving nearly every month. But what do we find where boilers are not used in an intelligent manner? How is it when the fires are drawn, the hot water blown out, and the cold water immediately run into them? The top is covered with brickwork, which retains the heat, while the lower part is cooled off by the water, which causes it to contract or shorten, and so it is not to be wondered at, when the seams must be caulked or the tubes expanded. Even when one

boiler of a battery is allowed to cool off, and is then filled with cold water, and the steam is turned into it from the other boilers, the effect is just as bad; yet this is often done in some places. In addition to all this and more that might be added, internal corrosion is getting in its work of weakening the shell, heads and tubes, and when at last the crystalized plates give out, or, in other words, an explosion occurs, we are told that the fireman was careless and let the water get low, or that some mysterious agency has been at work which is beyond the power of man to control, therefore nobody is to blame.

CHAPTER XLIII.

STEAM PIPE COVERINGS.—COMBUSTION OF FUEL.

If the question is asked as to what kind of a covering for steam pipes is the most efficient, it will be safe to reply that on general principles that covering which is the most porous, or in other words which allows the greatest quantity of air to be confined within its limits, will prove the best. As for the need of a covering of any kind that has been established by many experiments at different times and under various circumstances, and while in bare pipes the condensation is sometimes as great as half a pound of steam per hour per square foot of surface exposed, when the same pipes have been properly covered, this has been reduced to about one-eighth of a pound in the same time. Excessive condensation is not only expensive, but dangerous in some cases, for with high speed engines with small clearance, if the steam pipe is long, the water of condensation may not do harm of itself, although it will prove an annoyance, but if the boilers prime, even slightly, the two together may wreck the engine, when neither one alone would do it. The porous covering must be in turn covered with some substance that will not admit air through it, for while the air space is well known to be an excellent non-conductor of heat, still it must be remembered that this must be a dead-air space, for if it can circulate around the pipes its efficiency will be very low, or perhaps no better than none at all.

A covering made of straw, well protected from the outside air, has proved very efficient on account of the large air space that it formed, while a regular air space covering has proved a failure, because the air was allowed to circulate. A test was once made on a steam pipe of medium length, but which was rather large for the service required, to determine how much steam was condensed in its passage to the engine, and it was demonstrated that in cool weather fully 25 per cent. of the steam generated was lost, or condensed before it reached its destination. When the pipes were well protected this was reduced to about three per cent. Much attention has been given to this matter lately, and several first-class coverings are in the market, and may be procured at reasonable expense.

The air that we breathe is composed of oxygen and nitrogen, and the coal that the engineer is expected to make steam with is composed of carbon and hydrogen. We are of the opinion that some of the coal that we have to burn contains large quantities of other matter, and that some of the air that we breathe is diluted with foreign substances, but, however, these are the principal ingredients with which we have to deal. When we have a good fire in our boiler furnace there is a process of rapid oxidation going on which we call combustion. To carry out this process the carbon and hydrogen in the fuel combine with the oxygen in the air, and when we get the proper combination, or, in other words, the right qualities of these together, the result is perfect combustion. When we consider that the air is composed of oxygen 21 parts and nitrogen 79 parts by volume; or oxygen 77 parts and nitrogen 23 parts by weight, and when we consider that the

air that supplies our furnaces is supplied by volume, or, in other words, according to the space vacated by the products of combustion, we see that we must take in much that we do not need, in order to get what we do want. It is customary, however, to speak of the number of pounds of air that we need to burn a pound of coal, and we are told that average coal chemically consumes 10.7 pounds of air for each pound of combustible, but ordinarily a great deal more than this is used, as we practically never realize such results in practice. Again, when we consider that when oxygen and hydrogen are present in the right proportions for it, they combine to form water, instead of fire, and in that case more heat must be taken to evaporate this water, which, of course, is a loss. Another authority tells us that the minimum amount of air that will be needed for the best grades of coal is 12 pounds of air for each pound of coal as we find it, and as each pound of air at a temperature of 62° F. occupies 13.14 cubic feet of space, we see that each pound of coal needs $13.14 \times 12 = 157.68$ cubic feet of air when burned under the most favorable conditions, and these conditions are not present in an ordinary boiler furnace. On the contrary, many times the conditions are such that 24 pounds, or 315.36 cubic feet, are used for each pound of coal disposed of. From the above it will be seen that if the candidate for a license is asked how many pounds of air is disposed of for each pound of coal consumed, he should reply: "From 12 to 24 pounds, according to conditions."

CHAPTER XLIV.

BURSTING PRESSURE OF STEAM BOILERS, THICKNESS AND TENSILE STRENGTH OF BOILER PLATE.

In addition to what was said in Chapters 14, 15 and 16 concerning the strength of a boiler, the following will be found of value to the engineer. They were taken from standard works on steam engineering, but as such formula are given in a very condensed form, often without examples by way of illustration, the average engineer finds it rather hard work to understand them. The author needs no one to tell him this, for he knows it by experience, and if he can succeed in putting matter in more extended form, so that it may be more readily applied by his associates in the profession of steam engineering, one of his objects will be accomplished.

The following are given for determining the bursting pressure, thickness of plate, and tensile strength of plate, respectively:

$$\frac{4480\, t\, s}{d} = p$$

$$\frac{d\, p}{4480\, s} = t$$

$$\frac{d\, p}{4480\, t} = s$$

in which t=thickness of plate in decimals of an

inch, s=tensile strength of plate in tons per square inch, d=internal diameter in inches, and p=effective or bursting pressure per square inch. For illustration we will assume that we have a boiler 60 inches in diameter, made of iron plates .375 inch thick, whose tensile strength is 50,000 pounds per square inch of sectional area, and the longitudinal seams are double riveted and possess .70 of the strength of the solid plate. Applying the first formula we find that 4480×.375=1680. The value of S is to be stated in tons, and in the example it is given at 50,000 pounds, therefore to reduce it to tons we must divide by the number of pounds in an English or "long" ton, and 50,000÷2240=22.32. 1680×22.32=37,497.6.

In the formula the internal diameter of shell is to be taken, which of course is proper, but in the case of plates one-half inch thick, or less, it is customary to take the outside diameter, as by so doing the matter is much more simple, and the difference in the result is small, being but about one pound, for a safe working pressure in this case: 37497.6÷60=624.96 pounds, which is the value of p in the formula as applied to this example. This, it must be remembered, is the bursting pressure of a boiler made without riveted seams. As we have assumed that the longitudinal seams possess .70 of the strength of solid plate, we multiply 624.96 by .70 and our product is 437.472, and if we use 5 as a factor of safety we find that the safe working pressure is 87.5 pounds.

Referring to the second formula, and substituting for the symbols their values as determined above, gives us the following:

$$\frac{60 \times 624.96}{4480 \times 22.32} = .375 \text{ inch.}$$

which is the thickness of plate. Referring to the third formula, and substituting for the symbols their values, we have

$$\frac{60 \times 624.96}{4480 \times .375} = 22.32 \text{ tons per square inch.}$$

If we wish to know how far out the centre of the heads of this boiler may be bulged without exceeding the elastic limit, or, in other words, how far out of truth they may be sprung at the centre and still return to their original position when the pressure is removed, we may use either of the following formulæ, assuming that there are no braces or tubes to hold them in position:

$$\frac{r}{22} = b, \text{ or } \frac{d}{44} = b$$

in which r equals radius, d equals diameter and b equals bulging pressure. Assuming the heads to be of wrought iron one-half inch thick gives us the following:

$$\frac{30}{22} = 1.36 \text{ inches, or } \frac{60}{44} = 1.36 \text{ inches.}$$

If we wish to know how much pressure per square inch it will require to do this, we may use the following:

$$\frac{815\, t\, s}{d} = p$$

in which t = thickness of head in inches, s = tensile stress of the head in tons per square inch at the the elastic limit, d = diameter of the boiler, and p = pounds per square inch.

Applying this to the case in hand, and substituting for the symbols their values, and our formula becomes

$$\frac{815\times.5\times11.16}{60}=75.8$$

pounds. Using 5 as a factor of safety gives us 15.16 pounds as the safe working pressure of such a head without braces or stays. An explanation of the way that we obtain the value of S in this case is in order. As before noted, we take tensile strength of the iron at 50,000 pounds, or 22.32 tons, and as the elastic limit is one-half of this, we divide 22.32 by 2, and the quotient is 11.16 tons. As even numbers are often used in these calculations, the value of S is taken at 12 tons, which is proper here, whence the following formula is derived:

For an iron head

$$10{,}000\frac{t}{d}=p$$

and for a steel head

$$11{,}500\frac{t}{d}=p$$

in which t= thickness of head in inches, d=diameter of the boiler, and p=bulging pressure in pounds per square inch. Making use of the former, we have

$$10{,}000\frac{.5}{60}=83$$

pounds bulging pressure, or 16.6 pounds safe working pressure. This is a trifle more than we obtained by a preceding formula, but this increase is due to the fact that elastic limit is taken at 12 tons, which makes the tensile strength 24 tons or 53,760 pounds, instead of 50,000 pounds as we assumed. The terms "pounds" and "tons" are

both used here in order that the reader may fully understand both of them, but we prefer "pounds" wherever practicable, as it agrees with American practice, while the ton is more frequently used by the English engineer.

CPAPTER XLV.

FLAT, CONCAVE AND CONVEX BOILER HEADS.

If the head of the boiler that we took for illustration were made of steel instead of iron we would use the following formula:

$$11500\frac{t}{d}=p$$

in which the symbols refer to the same elements as in that given for an iron head. Applying this gives us

$$11500\frac{.5}{60}=95.45$$

pounds, and with a factor of safety of five, the result is 19.09 pounds. Suppose that this boiler is equipped with a dome 24 inches in diameter, and this dome has a cast iron head two inches thick. If we wish to know how much this will bulge at the centre, without exceeding the elastic limit, we can use the following formula to determine it:

$$\frac{d}{44}=B$$

and when we wish to determine the pressure that will do this we use the following:

$$4000\frac{t}{d}=p$$

In which d=diameter of head, B=deflection at

the centre, within the elastic limit, t=thickness of head in inches, and p=pressure in pounds per square inch. Applying these we have

$$\frac{24}{44}=.545 \text{ inch, and } 4000\frac{2}{24}=332$$

pounds bulging pressure, or 66.4 pounds safe working pressure. According to the above this head is much weaker than other parts of the boiler, but there are several things to be taken in consideration which effect the result. The formula is based on the assumption that cast iron has an elastic strength of five tons per square inch of sectional area, as being an average, but while some very poor specimens have shown even less strength than this, other good specimens possess double this strength, and exceptionally good ones have demonstrated that it requires 17 tons to break them, which would make the allowable pressure more than three times that given by the formula.

Remelting the iron improves its quality, and without doubt none but the very best cast iron that can be procured should ever be used in boiler construction. This material possesses but very little elasticity, so that the tensile strength is but little more than the elastic strength. Usually the steam nozzle is cast on the dome head, and while the hole tends to make the part weak, still the raised portion of the nozzle reinforces it, and in a great measure counteracts its bad effects. If it is necessary to drill holes in such heads to make minor steam connections they should be located as far apart as possible.

The foregoing formulas refer to flat heads, and ordinarily these are what the engineer has to deal with, but it may be well for him to understand

how to calculate the pressure that a spherical head will stand, or in other words a head made in convex form, or in the engineer's vernacular, "a bulged head." Such a head cannot very well be used for a tube sheet on a boiler, but there are other places where such heads are used. Their strength may be determined by use of this formula

$$\frac{8960\,t\,s}{\dfrac{r^2}{v}+v.}=p.$$

in which p=pressure in pounds per square inch, t=thickness of the head in inches, s=elastic strength of head in tons per square inch, r= radius of the head, and v=versed sign or rise of the head in inches.

Assuming elements of the preceding case, except that the head is bulged out six inches at the centre and made spherical in form by the boiler makers, instead of being flat, and substituting for the symbols their values, our formula becomes

$$\frac{8960\times .5\times 11.16}{\dfrac{30^2}{6}+6}=320$$

pounds, elastic limit, or 64 pounds safe working pressure.

Having explained the way to read many of the formula which appear in the preceding chapters of this book, I have considered it necessary to explain those in this chapter further than to give the figures which illustrate their application, but

as the one just given looks rather formidable, a full explanation may be advisable.

The number 8960 is to be multiplied by the thickness of the head in inches or parts of an inch, and the product by the elastic strength in tons per square inch, and this we may term our first product. The radius, or one-half of the diameter of the head is to be multiplied by itself and the product divided by the rise of the head in inches. To this quotient add the rise of the head and the sum will be our second product. Divide the first by the second and the quotient will be the elastic limit in pounds per square inch.

Thus we see that by giving the head a spherical form, the rise of the segment being six inches, we have increased the safe working pressure from 15.16 pounds in the flat head to 64 pounds in the spherical one. A careful study of these formulas will show that in calculating the safe working pressure of the shell, the ultimate tensile strength of the iron is used, while for the heads the elastic strength is taken, but if we wish to make a comparison of the strength of the shell of a boiler, made without seam or weld, and of a spherical head, we must take the same quantity for both, which may be stated at 24 tons tensile strength.

In this case both shell and head must be of the same thickness in order to make the comparison proper. If a 60-inch boiler shell is made of iron one-half inch thick we find that it will take $4480 \times .5 \times 24 \div 60 = 896$ pounds to burst it, and if the head is spherical, the rise of it being eight inches, it will require $8960 \times .5 \times 24 = 107520$; $30 \times 30 \div 8 + 8 = 120.5$; $107520 \div 120.5 = 892$ pounds to rupture it, therefore if the rise of the spherical head is made one-eighth of the diameter of the shell, or one-

fourth of its radius, approximately, both head and shell will possess equal strength. Care should be taken to avoid confounding the tensile strength of the iron in these calculations with its elastic limit.

A careful study of these problems will show that, if the wrought iron head of a plain cylinder boiler should be made concave, it would be much stronger than either a flat or a convex one, for the reason that the pressure tends to compress the iron, and the same authority that tells us that the ultimate tensile strength of iron in tension is about 24 tons, also tells us that "the resistance to compression is an indefinite quantity."

From this it would appear to the casual reader that the strength of a concave head is so great as to be almost unknown, but when we consider that the concave head will not retain its form under great pressure, and if it did, it would bring an excesive stress to bear on that part of the shell which immediately surrounds it, we see that there are modifying conditions present.

CHAPTER XLVI.

WROUGHT IRON, STEEL AND CAST IRON BOILER HEADS.

It may be well to give a separate formula to determine the elastic limit of strength of spherical boiler heads when made of wrought iron, steel or cast iron. They are as follows:

$$\frac{108{,}000\,t}{\frac{r^2}{v}+v}=p$$

for a wrought iron head.

$$\frac{125{,}000\,t}{\frac{r^2}{v}+v}=p$$

for a steel head, and

$$\frac{45{,}000}{\frac{r^2}{v}+v}=p$$

for a cast iron head. In the above formulas the elastic limit of the metal is taken at 12 tons for wrought iron, 14 tons for steel and 5 tons for cast iron, and the explanation found in Chapter 45 will apply here. In the 60 inch wrought iron boiler that we have taken for illustration in work-

ing out and applying these rules and formula there are some flat surfaces which will need to be braced, as for instance the heads above and below the tubes, and when these are not strengthened except by the braces, the following formula may be safely used:

$$\frac{407\, t\, s}{a} = p$$

in which t = thickness of the plate in inches, s = strength of the plate in tons per square inch at the elastic limit, p = maximum elastic pressure in pounds per square inch, and d = clear distance apart of the braces. By the latter is meant the distance from the outside of one brace to the outside of another, and not from centre to centre, as for instance if a section of plate is laid out in squares and braces one inch in diameter are given a pitch of eight inches, or it is eight inches from centre to centre of them, then the clear distance apart is seven inches.

If the plate is not laid out in squares, but in rectangles or irregular forms, then the greatest clear distance apart must be taken for the calculation, in order to be on the safe side. Let us apply this formula to the boiler head taken for illustration, which is one-half inch thick, and assume that the braces are placed in squares eight inches from centre to centre, making the value of d seven inches. It will be remembered that we have taken the value of S at 11.16 tons for a reason already explained. Our formula thus becomes

$$\frac{407 \times .5 \times 11.16}{7} = 324.437$$

pounds pressure at the elastic limit. If we adopt a factor of safety of 5, we find that our safe work-

ing pressure is 64.887 pounds. We shall show presently how this head is to be made strong enough to stand a higher pressuse than this, and at the same time would state that we are well aware that it is not practical to lay out this segment of a bóiler head in equal shares of eight inches, but the reader can readily see how these rules can be applied to the water legs of locomotive and marine boilers or for any similar work.

If we already have the values of p, t and s, and wish to determine the value of d, our formulr becomes

$$\frac{407\ t\ s}{p} = d$$

and its application

$$\frac{407 \times .5 \times 11.16}{324.437} = 7 \text{ inches.}$$

If we already have the values of p, d and s, and wish to ascertain the value of t, the formula becomes

$$\frac{p\ d}{407\ s} = t$$

and its application

$$\frac{324.437 \times 7}{407 \times 11.16} = .5 \text{ inch.}$$

If we already have the values of p, d and t, and wise to ascertain the value of s, the formula becomes

$$\frac{p\ d}{407\ t} = s$$

and its application

$$\frac{324.437 \times 7}{407 \times .5} = 11.16 \text{ tons.}$$

If the valve of s is taken at 12 tons, we may use the following, which are somewhat shorter:

$$\frac{5000\,t}{d} = p. \quad \frac{p\,d}{5000} = t. \quad \frac{5000\,t}{p} = d.$$

If the head is made of steel we may take the value of s at 14 tons per square inch, and the following may be used

$$\frac{5700\,t}{d} = p. \quad \frac{p\,d}{5700} = t. \quad \frac{5700\,t}{p} = d.$$

But even after we have figured out all of the above examples, it does not tell us whether the brace is strong enough to withstand the strain that will come upon it when our boiler is put into service, and for this purpose we must use the following formula, remembering that the maximum elastic pressure is taken at 324.437 pounds, or rather it was so determined by one of the foregoing rules, and it may be well to remind the reader that this formula is to determine the diameter of a brace that will stand just the same pressure that the plate will, under like conditions:

$$.024\sqrt{\frac{P\,P'\,p}{S}} = d$$

In which P = pitch of stays or braces between centres longitudinally, or in other words lengthwise, P' = pitch of braces, transversely or crosswise, p = maximum elastic pressure in pounds per square inch, S = elastic strength or iron in the braces, in tons per square inch, which must correspond to the elastic strength of the plate, and

d = diameter of brace. Our first business is to make the reading of the formula plain, and it is as follows: Multiply P. P' and p together and divide the product obtained by S.

Ascertain the square root of the quotient and multiply it by .024; substituting for the symbols their values the formula becomes

$$\frac{.024\sqrt{8\times8\times324.437}}{11.16}=1 \text{ inch}$$

which corresponds to the diameter assumed in order to illustrate preceding formulas.

Again, when the value of S is taken at 12 tons, the formula becomes $.0069\sqrt{P\,P\,p}=d$, and if S is equal to 14 tons, as in the case of steel being used, it is $.0064\sqrt{P\,P'\,p}=d$.

If the brace of stay bolt has a thread cut on it, the diameter must be taken at the base of the thread.

CHAPTER XLVII.

BRACING STEAM BOILER HEADS.

It now becomes necessary for us to give the heads of this boiler a little more attention, in order to determine how we may make that portion of them which is above the tubes strong enough to be safe under the working pressure of 87.5 pounds, for we have demonstrated that even with braces placed eight inches apart the heads are only fitted to carry about 65 pounds, and as it is not practical to have so many as this arrangement calls for we must resort to some other means for stiffening this important part, until it will be perfectly safe under the working pressure that we have adopted. The shape of the part to be braced is in the form of the segment of a circle, and the first question to be settled is as to what portion of this segment is already stiffened by the flange of the head and by the tubes.

If we were to provide braces for the whole of this part of the head above the tubes, we should be on the safe side, and many authorities on this subject evidently intend for this to be done, as they ignore the matter altogether, while others simply state that the flange imparts strength to an arbitrary number of inches from the diameter of it, as was done in Chapter 16 of this book, which statement was intended to be on the safe side, but a rule will now be given which may be applied to any boiler, and which we shall show is

based on correct principles. The distance that the flange imparts strength to is due to the thickness of the head, and as a matter of course, to the pressure to be carried, and these points are also what determines the pitch of the stays or braces, and as we have not yet shown what this pitch would be in the case that we have assumed, we will now do so.

Our safe working pressure is 87.5 pounds and factor of safety, 5, therefore the pressure which equals the elastic limit is 87.5×5=437.5 pounds. The safe working pressure for the shell is based on the tensile strength, but the safe working pressure of the head is based on the elastic limit. Applying a formula already given to determine the clear distance apart of the stays we find that

$$\frac{407 \times .5 \times 11.16}{437.5} = 5.19 \text{ inches.}$$

or 6.19 from centre to centre, which we will call 6 for convenience sake, as it will not materially alter the result. This means that each stay will support the pressure on a space of six inches square, or in other words if we marked a space six inches square and put the stay in the center of it, the plate will be well supported to this extent. This being the case we may safely regard the fastenings of the head to the shell as a stay which is able to support a space of three inches wide each side of it, or in other words, the diameter of the circle to be braced will thus be reduced by three inches on each side, or six inches in all. From the above we derive the following formula for the distance supported by the flange:

$$\frac{407\, t\, s}{P\ 2} + \text{B} = d$$

in which t= thickness of the plate, s= strength of plate at elastic limit in tons per square inch, P= maximum elastic pressure in pounds per square inch, B= one-half diameter of brace and d= distance from circumference. Applying this to our our boiler, we find that

407×.5×11.16=2271.060; 437.5×2=875; 2271.060 ÷875=2.59+.5=3.09 inches. We may also regard the top row of tubes as braces, and as we have seen that the plate is strengthened for a distance of three inches from the centre of brace, or 2.5 inches from the outside of it, so we safely assume that a space 2½ inches wide above the upper row of tubes is provided for by the tubes.

The outside diameter of our boiler is 60 inches, therefore the head is 59 inches in diameter and the distance from the centre of it to the upper part of the shell is 29.5 inches. If we locate the upper row of tubes five inches above centre of head, and allow 2.5 inches more for strength imparted by the tubes, we find that the base of segment will be 7.5 inches above the centre of circle, and subtracting six from 59 for reasons already explained, the radius of the circle, which is to form the arc of the segment is found to be 26.5 inches, and this is the surface which must be held by braces.

In other words, draw a circle 53 inches in diameter, and then divide it into two unequal parts by a straight line 7.5 inches drawn above the centre of it. The smaller part represents the area to be braced, and in order to reduce the number of braces necessary, we will rivet on pieces of T iron, and while these are often placed radially, we propose to locate them in a horizontal position, and calculate their strength in this position, in order to illustrate a plan that is sometimes adopted in

such cases. These irons will be considered as flanged beams of unsymmetrical proportions, and their strength given accordingly. It will also be necessary to ascertain the exact area of the space to be braced, and multiply it by the pressure to be carried, that we may know how much of a load to provide for. There are a variety of rules for this purpose, some of which require the use of a table to use them, but we will illustrate one which any engineer can use at his discretion, the only articles necessary being a clean spot on the engine room floor on which to draw a circle, a wire nail for the centre of it, a piece of wire with a loop at each end of it of a proper length to represent the radius of the circle, a pencil, a straight-edged ruler and a pair of compasses.

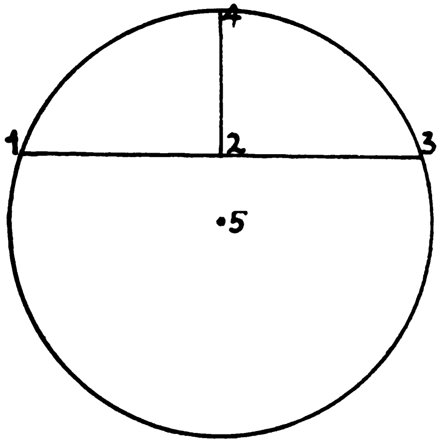
Having drawn the circle and laid out the segment, we will proceed in our next chapter to ascertain the area of it.

CHAPTER XLVIII.

AREA OF SEGMENTS OF CIRCLES.

Rule to determine the area of the segment of a circle when it is less than a semi-circle—Multiply one-half of the arc by the radius of the circle, and call the result the first product. Multiply one-half the length of the chord by the distance from

Fig 10

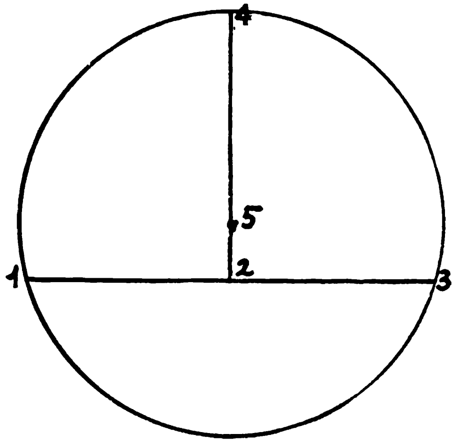


the centre of the circle to the centre of the chord, and call the result the second product. From the first product subtract the second and the remainder will be the area of the segment. A few words by way of explanation of terms used may not be

out of place. Segment means a part cut off or divided, so in this case it means a part of a circle. Arc is a contraction of the word arch, and as used here it refers to the curved line which forms the upper part of the segment.

The chord is the straight line which forms the remainder of the segment, and connects the two ends of the arc. To illustrate the foregoing rule and apply to the case in hand we refer to Fig. 10, which is a circle supposed to be 53 inches in diameter, the upper part of which is the segment that we wish to ascertaiu the area of. The chord is drawn 7 1-2 inches above the centre for reasons already explained. The distance from 1 to 4 is

FIG 11



34 7-32ds or 34.21875 inches, and the radius, or one-half of the diameter of the circle, is 26.5 inches; 34.21875 × 26.5 = 906.79 square inches, which is our first product. The distance from 1 to 2 is 25 3-8 or 25.375 inches, and from 2 to 5 is

7.5 inches; 25.375×7.5=190.31 square inches, which is our second product; 906.79−190.31 716.48 square inches, which is the area of the segment.

While we are on the subject we will give a rule for determining the area of a segment of a circle —when it is larger than a semi-circle.

Multiply one-half the length of the arc by the radius of the circle and call the result the first product. Multiply one-half the length of the chord by the distance from centre of circle to centre of chord and call the result the second product, Add the two products and their sum will be the area of the segment. To illustrate we refer to Fig. 11, which represents a circle 53 inches in diameter, and the upper or larger part of it is the segment that we desire to know the area of. This corresponds to the lower part of Fig. 10. Applying the rule we find that the distance from 1 to 4 is 49 7-32ds or 49.21875 inches.

The radius of the circle is 26.5 inches, and 49.21875×26.5=1304.29 square inches, which is our first product. The distance from 1 to 2 is 25 3-8 or 25.375 inches, and from 2 to 5 it is 7.5 inches; 25.375×7.5=190.31 square inches; 1304.29 +190.31=1494.6 square inches, which is the area of the segment. If we add the two areas together we will have the area of the whole circle, it is evident, and 716.48+1494.6=2211.08 square inches. If we desire to get the area of the whole circle directly we would have squared the diameter and multiplied by .7854, and 53×53×.7854=2206.18, and as the result obtained indirectly varies but 4.9 square inches from this, the difference is small when we consider the total number of inches concerned in the calculation.

In measuring the distances on curved lines it is

well to take the dividers or compasses to do it with, as accuracy is thus secured, and would also suggest that the whole of the arc be measured and the sum divided by two rather than to draw a vertical line from the centre of the chord to centre of arc, or to what is supposed to be the centre, for unless very great care is exercised an error will be introduced here. As we have demonstrated, the area above tubes to be strengthened by tee irons and braces contains 716.48 square inches, and as we proposed to carry 87.5 pounds on each square inch, 716.48×87.5=62,692 pounds to be supported.

If we were to put in braces of round iron 1 1-8 inches in diameter, allowing 6000 pounds to each square inch of sectional area in the brace, we should have 62,692÷6000=10.4, or, say 11 braces. As we have 716.48 square inches to support and 11 braces to do it with, to ascertain the number of square inches each one would be called upon to support, we may divide 716.48 by 11, and our answer is 65.13, the square root of which is 8.062, or practically the space supported by each brace will be 8 inches square. But, as before stated, we do not propose to put in so many braces, but will introduce tee irons laid horizontally, and first we will consider them as being supported at the ends only, although there are conditions present which modify this, and which tend to make the structure stronger than ever.

This head will support a light pressure without either braces or tee irons, but this fact is usually ignored in such calculations, and as it is an error on the safe side, if an error at all, it is not to be condemned. The radius of this head is 29.5 inches, but as it is supported for a distance of 7 1-2 inches above the centre by the tubes and for

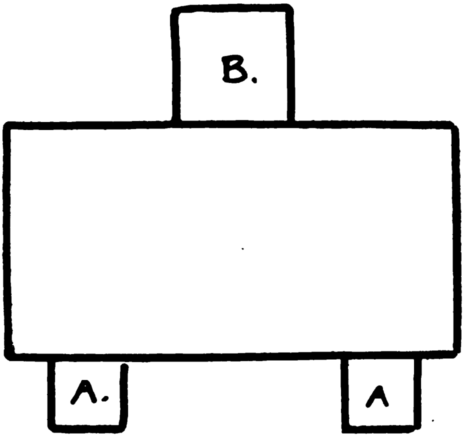
a distance of three inches from the shell by the flange, the height that we have to provide for is reduced to 19 inches, and we propose to use tee iron of a uniform thickness of one-half inch and a depth of four inches, or in other words $4 \times \frac{1}{2}$-inch tee irons.

CHAPTER XLIX.

STRENGTH OF TEE IRONS.

In calculating the strength of tee irons, there are several points to be taken into consideration, and, in fact, it is a subject that bristles with points. To begin at the foundation of the matter we refer the reader to Fig. 12. It represents a

FIG. 12.

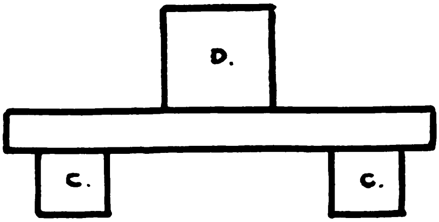


piece of iron one-half inch thick, and 3½ inches wide. It rests upon the blocks A A at each end, and supports the weight B laid upon its centre. In Fig. 13 we have a piece of iron one-half inch thick and four inches wide.

It also rests upon blocks C C at each end, while the weight D is laid upon its centre. The point we wish to illustrate by these figures is that the iron that is set up edgewise in Fig. 12 will support a much greater weight at its centre than will the iron which is laid down flatwise, as in Fig. 13, even if they were both the same size. The comparative strength of irons used in this way has been determined by experiment and found to be as 1 is to 1.73. In these calculations the iron shown in Fig. 12 will be referred to as the web, and that in Fig. 13 as the flange, therefore if the strength of the flange be taken as 1 then the strength of the web will be 1.73, for if the flange will support 1000 pounds, then the web will support $1.73 \times 1000 = 1730$ pounds.

Again referring to Fig. 12 we find that when the weight is put upon it the lower part has to withstand a direct pull, or in other words is in tension, while the upper part of it is pressed

FIG. 13.



together, or is in compression. There is a point in this bar of iron that is neither in tension nor compression, and as the bar is the same size throughout, or in other words is symmetrical in section, this point is found to be at the centre of

the bar, measured vertically, and is called its neutral axis. If one edge of this bar of iron were to be made thicker than the other, then the neutral axis will not be at the centre, but will be moved toward the thickest edge, therefore when we put the two bars of iron shown in Figs. 12 and 13 together we have a piece of tee iron, the width of both flange and web being four inches, when considered singly, and their thickness, one-half inch, or in other words we have a tee iron four by one-half inches shown in section in Fig. 14.

The reader will notice at once that the neutral axis of this iron is much below its centre, as the

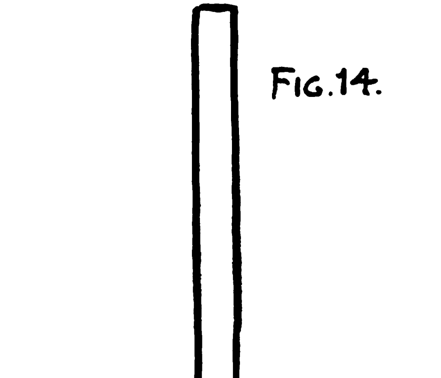


FIG. 14.

lower edge is thicker than the upper, but how much below the centre is the next problem to be solved, and for this purpose we refer the reader to Fig. 15. By extending the web down across the flange, as shown by the dotted lines at E E, we have a space, or part of the flange, at either end, which is 1.75 inches long, and it is these parts

which we must reduce to their equivalent in the web, and, as before stated, the difference between them is as 1 is to 1.73; therefore it becomes a problem in proportion, for 1.73:1 : : 1.75:1, or in short, 1.75÷1.73=1, which is the required distance in inches. So we draw the dotted line at F F, showing the reduction in the flange.

As this flange is all in tension, the question naturally arises as to why it is not proper to take the whole of it into consideration, but it should

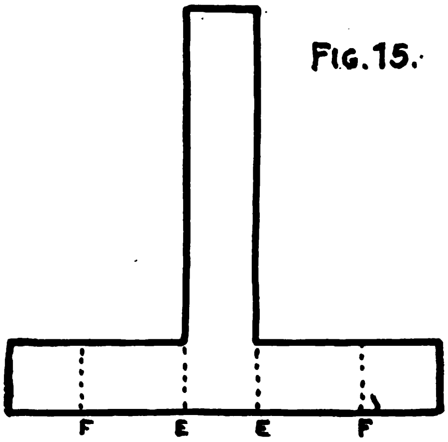


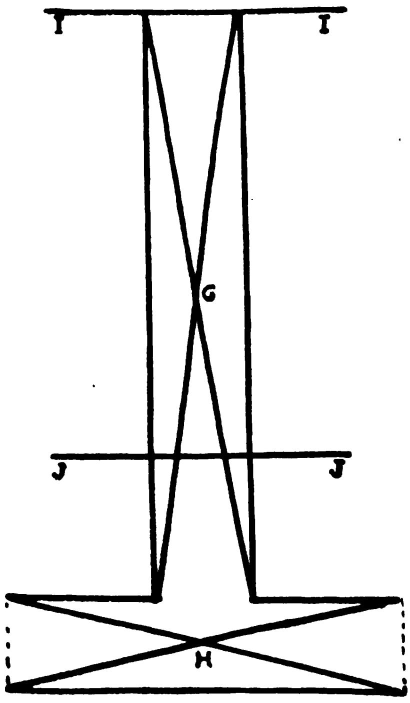
Fig. 15.

be remembered that the flange if taken by itself will bend much easier than the web, and as the tee iron must remain rigid, it is necessary to make this reduction to show its equivalent to the web.

In other words that part of the flange which is between the dotted line F F if set up edgewise would be equivalent to the whole flange taken as it is. The next step is to ascertain the centre of gravity for the web above the flange, and also for the reduced section of the flange. This is done

as shown in Fig. 16. For the web it is at G, and for the flange at H. We will now draw a line across the top, as shown at I I, which we will call a datum line, because from it we must take some measurements.

FIG 16



The next step is to ascertain the areas of the web and flange, respectively, and multiply them by the distance of their centres of gravity from the datum line. The area of the web is 3.5×.5= 1.75 square inches, and its centre of gravity is 1.75 inches from the datum line; 1.75×1.75=3.06. The area of the flange is 2.5×.5=1.25 square

inches, and its centre of gravity is 3.75 inches from the datum line; 1.24×3.75=4.68. These two results are to be added together and 3.06+4.68=7.74, which is to be divided by the sum of the areas of the web and flange, and 1.75+1.25=3; 7.74÷3=2.58 inches, which is the distance of the neutral axis of the complete four-inch tee iron from the datum line. This axis is shown by the line J J and must be considered as a fixed point, from which other measurements are to be taken and other calculations made, the first of which is to determine the tensile resistance of that portion of the four inch tee iron which is below the neutral axis, but from the foregoing it is plain that the reduced section of the flange must be taken, instead of its whole section.

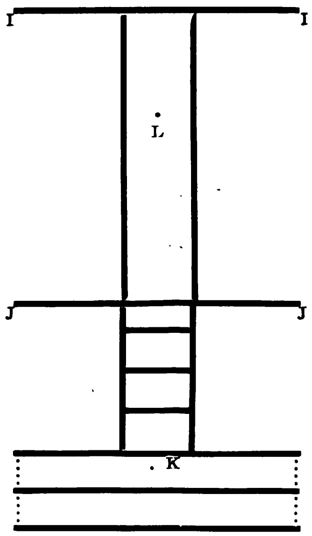
A moment's consideration will also show us that as the neutral axis is to be regarded as a fixed point, that portion of the iron which is directly beneath it, say the first quarter-inch in depth, is not so effective as the lowest quarter-inch of the web, or of a like amount of iron at the extreme lower part of the flange, for in a certain sense this tee iron, Fig. 16, may be regarded as a lever of the first kind, in which the neutral axis J J represents the fulcrum, the datum line I I the weight to be raised, and the lower part of the flange is where the power is applied. By this illustration it will be seen that a square inch of the iron at the lower part of the flange is more effective than a square inch immediately under the neutral axis, and also that each square inch of iron or fraction thereof, is effective according to its distance from the fulcrum, that is the natural axis. We will divide the part in tension into sections and multiply the area of each section by the distance from the centre of it to the neutral axis.

CHAPTER L.

COMPRESSIVE AND TENSILE STRENGTH OF SECTIONS OF TEE IRONS.

In Fig. 17 we have the reduced section of the four-inch tee iron, divided into small sections, each one quarter of an inch wide beginning at the

FIG. 17.



bottom and extending up to the neutral axis. The lower one has an area of $2.5 \times .25 = .625$ square inch, the next one above it is the same, the next is

$.25 \times .5 = .125$ square inch, the next is the same, and also the next, but the one next to the neutral axis is smaller, being $.17 \times .5 = .085$ square inch. Their respective distances from their centres to the neutral axis are, first, 1.295 inches; second, 1.045 inches; third, .795 inch; fourth, .545 inch; fifth, .295 inch; sixth, .085 inch.

Multiplying these together as described in the closing paragraph of Chapter 49, we have

$$\begin{aligned} .625 \times 1.295 &= .809 \\ .625 \times 1.045 &= .653 \\ .125 \times \ .795 &= .099 \\ .125 \times \ .545 &= .068 \\ .125 \times \ .295 &= .036 \\ .085 \times \ .085 &= .007 \end{aligned}$$

These products are to be added together, and their sum is 1.672, which is to be divided by the distance from the neutral axis to the extreme lower part of section, which in this case is $4 - 2.58 = 1.42$ inches, and $1.672 \div 1.42 = 1.17$. This quotient is to be multiplied by the constant number 1.73, and the product by the total tensile strength of the iron per square inch, which we assume at 50,000 pounds. $1.17 \times 1.73 \times 50{,}000 = 101{,}205$ pounds which is the strength of the section below the neutral axis. Not the strength per square inch, but the total strength, as we have already taken into account the area.

We must next locate the centre of the tensile stress of this four-inch tee iron, which is the centre of tensile stress of the reduced section shown in Fig. 17, or rather that part of it which is below the neutral axis J. J. To do this we must multiply the area of each strip by the square of the distance from its centre to its neutral axis. Add these products together and divide the sum so obtained by the sum of the first products (in this

case 1.672 inches), and the quotient will be the distance from the neutral axis to the centre of the tensile stress. The first or lower one has an arc of .625 square inch, and its distance is 1.295 inch, therefore in carrying out the above directions we have .625×1.295×1.295=1.048. In treating the second one the same way we have .625×1.045×1.045=.682. In the case of the third it is .125×.795×.795=.079. The fourth gives us .125×.545×.545=.037. The fifth is .125×.295×.295=.010, and the sixth .085×.085×.085=.0006. Add these products together and their sum is 1.8566. The sum of the first products is 1.672, therefore 1.8566÷1.672=1.11 inches, which is the distance from the neutral axis to the centre of the tensile stress. See the point K in Fig. 17.

Our next move is to find the centre of the compressive resistance, namely of that portion of the web which is above the neutral axis J. J. The same process which we used to find the centre of the tensile stress K may also be used here, but as this part is a rectangle we know that this point is at two-thirds of its height, and two-thirds of 2.58 inches is 1.72 inches, therefore the centre of the compressive resistance is 1.72 inches above the neutral axis See point L, Fig. 17. If for any reason it is desired to find the total compressive resistance (which is not required in this case), it must be remembered that while according to a general law the tensile and compressive resistance is the same in a section which is symmetrical, still in one which is not symmetrical, as in this case, the compressive resistance is greater in direct proportion or ratio, as the difference in the distance from the neutral axis to the centres of each resistance. Here it would be 1.11 : 1 : : 1.72 : 1.55, the latter denoting the excess of strength of the

compressive resistance, owing to its greater distance from the neutral axis. All that concerns this example in this connection is to see that the part in compression is strong enough to prevent buckling when the stress is put upon it, and as we have shown that the compressive strength is 1.55 times the tensile strength this point is well covered.

We have refrained from giving any rule or formula for determining the foregoing points in this calculation, as they would necessitate the use of many words and phrases which would need explanation, in following out our plan of endeavoring to make each point clear as we proceed, so that we have deemed an illustration of the way to proceed to be sufficient, as it can be applied to any beam of similar section by substituting the proper sizes. We now give a general formula, in continuing the foregoing illustration:

$$\frac{S\,4\,D}{L} = W$$

in which S = total tensile resistance of the section below the neutral axis under given conditions in pounds, D = the vertical distance apart of the centres of tension and compression in inches, L = the span in inches, and W = the breaking weight in pounds.

If the value of W is required in tons, then the value of S must be stated in tons. Substituting for the symbols their values our formula becomes

$$\frac{101205 \times 4 \times 2.83}{12} = 95470 \text{ pounds.}$$

CHAPTER LI.

CALCULATING THE LOAD ON TEE IRONS.

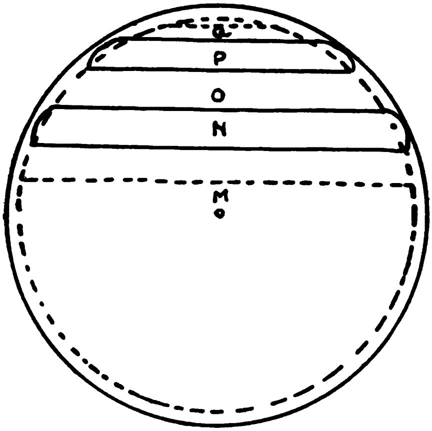
In illustrating the use of the preceding formula we have explained the way to determine the value of all the factors except that of L, which we have stated as being 12. It is now in order to tell how that is obtained, and for this purpose we will refer the reader to Fig. 18. The outside circle in full lines represents the head of the boiler that we have under consideration, which is 59 inches in diameter. The circle in dotted lines represents that part of the head which is supported by the flange, and its diameter is 53 inches. The dotted line M represents the top of the upper row of tubes, and is five inches above the centre of the circle.

The N represents a tee iron four inches wide. It is located five inches above the upper row of tubes, because the tubes are assumed to impart stiffness to 2½ inches of the head above them, and the tee iron will consequently stiffen another space 2½ inches wide, and this five-inch space is well within the limits of what the head will safely bear of its own thickness, as previously explained. The O represents another space five inches wide, and P is another four-inch tee iron, while Q is a space 2½ inches wide, which is supported by the tee iron. By adding together the above, it will be seen that while the radius of this circle is 26.5 inches, 25.5 inches are here provided for, leaving a small segment only one inch in height out of the

calculation, but as its area is small, and our factor of safety in calculating on the other elements is large, this will not materially affect the result.

Let us now see how much pressure the tee iron N will be called upon to support. By drawing a line through its centre extending from side to side of the circle represented by the dotted lines, we find that the average length of this line is 47.5 inches, which we will call 48 inches for convenience in calculating. The space that it has to support is nine inches wide, as already explained, therefore 48×9=432 square inches, and as we are to carry 87.5 pounds pressure, 432×87.5=37,800 pounds. We will make our braces of round iron

FIG 18



1½ inches in diameter, the area of which is 1.767 square inches, and as we are not to exceed a pull of 6000 pounds per square inch on our braces, 1.767×6000=10,602 pounds, which is the limit for each brace; 37,800÷10,602=3.56, so that we shall

need four braces. By putting one brace six inches each side of the centre of this iron, and another one 12 inches from each of these, we find that they are evenly distributed, and that the greatest span or distance between supports is 12 inches. This is how we determined the value of L in the formula.

In puttiug in these braces it would be well to run the two nearest the centre from head to head, and the two outside ones from head to shell. In determining the space to be supported by the upper tee iron, we find that its average length is about 32.25 inches. By locating one brace in the centre of this iron, and one each side of it at a distance of 11 inches, our load will be properly distributed, and as the span here is less than in the case of the lower one, we are on the safe side. As the space to be supported by the tee iron is 32.25 inches long and 9 inches wide, it contains 290.25 square inches, and the total pressure on it is 25,396 pounds. As each brace is safe under a pull of more than 10,000 pounds, the margin is large enough to cover every contingency.

As shown in Chapter 50, this span of 12 inches has an ultimate strength of 95,470 pounds. Taking one-half of this as the elastic limit, and using a factor of safety of five, as before, we find the safe load to be 9547 pounds. Let us see whether our working load exceeds this or not. The span is 12 inches long, and it supports a space nine inches wide; $9 \times 12 = 108$ square inches, and multiplying by the pressure gives us $108 \times 87.5 = 9450$, which is a trifle less than the safe load. With the upper tee iron the working load will be less than this, showing that we are still on the safe side.

If the lower row of tubes are so low down that only a hand hole can be put in each head, below

the tubes, then no braces will be needed here, but if a manhole is put in the front head, then braces will be needed according to conditions, but usually two are enough, and they should extend from head to head, as it is poor practice to rivet a brace to the shell directly over the fire,

It may be the opinion of some readers that such a boiler is safe under a much higher pressure than this, and their opinion may be based on the fact that they know of similar boilers that are doing it, but what does that signify? A certain gentleman wanted to hire a coachman, and advertising the fact, was soon called upon by several applicants for the position. He submitted to each one in turn the following question: "Suppose that I engage you to drive for me, and while riding with my wife and family we should come to the vicinity of a precipice. How near could you drive to the edge of it and escape accident?"

The first one said that he could approach within a yard of it and get away safely. He was told that if his services were required he would be notified. The second one said that he could drive within a foot of it and yet be safe. He received the same reply as the first one. The third one said that he was such a complete master of his business that he could cause the carriage wheels to roll within one inch of the edge and still not go over it. He received the same reply as his predecessors. The fourth one said that he would keep just as far away from it as circumstances would allow. He was engaged at once.

CHAPTER LII.

SAFE WORKING AND COLLAPSING PRESSURE OF TUBES AND FLUES.

In some of the works on steam boilers there is very little said concerning the collapsing pressure of tubes and flues. This may be partly accounted for by the fact that in the ordinary tubular boiler, with three or four-inch tubes, the collapsing pressure of them is so much greater than the bursting pressure of the shell and heads, that they (the tubes) are practically out of the question. As an illustration of the truth of the above statement, we introduce the following formula:

$$\frac{4480\ S\ T}{D}=P$$

in which S=collapsing pressure in tons per square inch of section of metal, T=thickness of tube in inches and D=the external diameter in inches. The value of S is obtained by the following rule: From 25 subtract $2\frac{2}{3}$ times the diameter of the tube. For a 3 inch tube it will be $2\frac{2}{3}\times3=8$ and $25-8=17$ tons. For a 4 inch tube it will be $2\frac{2}{3}\times4=10.64$ and $25-10.64=14.36$ tons.

We will apply this to the case of a 4 inch tube .125 inch thick and when we have substituted for the symbols their values, our formula becomes

$$\frac{4480\times14.36\times.125}{4}=2010$$

pounds collapsing pressure. Or the following may be used as it is at times more convenient.

$$T\left(\frac{112,000}{D}-12,000\right)=P$$

By this we mean that if 112,000 is divided by the diameter of the tube and 12,000 subtracted from the quotient, then by multiplying the remainder by the thickness of the tube, the product will be the collapsing presure in pounds. Applying this to the four inch tube as before we have

$$.125\times\left(\frac{112,000}{4}-12,000\right)=2000$$

pounds collapsing pressure. This is practically the same as with the preceding formula. Using a factor of safety of 10, which corresponds to a factor of five when based on the elastic limit, the safe working pressure of this tube is 200 pounds, and if it were a three-inch tube it would be 316 pounds. The foregoing rules apply to small round tubes not more than 18 feet long, but not to large flues.

The United States Treasury rules for finding the strength of lap-welded flues, when the diameter is not less than seven inches or more than 16 inches, may be expressed by the following formula:

$$\frac{T\,4400}{R}=P$$

in which T=thickness of flue, R=radius of flue in inches, and P=the pressure that may safely be carried. This, however, is subject to modifications, one of which is that the length must not exceed 18 feet, for when they are longer than this a reduction in the pressure carried must be made

amounting to three pounds for each foot or fraction thereof over 18 feet. When the pressure is between 60 and 120 pounds, the flue must be made in lengths not to exceed five feet, and riveted together in a substantial manner, or corrugated, the effect of which is practically the same. If the flue is more than 16 inches and less than 40 inches in diameter, the same formula may be used, except that instead of using the constant number 4400 we must substitute the number 2840.

Applying the formula to a flue 14 inches in diameter and one-quarter of an inch thick, and substituting for the symbols their values, our formula becomes

$$\frac{.25 \times 4400}{7} = 157 \text{ pounds}$$

for a length of 18 feet or less, but if it were 22 feet long, the pressure would be 157−12=145 pounds.

Applying the latter to a flue 24 inches in diameter, less than 18 feet long, and three-eighths of an inch thick, the pressure would be

$$\frac{.375 \times 2840}{12} = 88.7 \text{ pounds.}$$

The rule given by the same authority to determine the thickness of a flue for a given pressure, is expressed by the following formula:

$$\frac{R\,P}{4400} = T$$

the letters denoting the same factors as in the two previous ones.

Applying this to the14-inch flue we have

$$\frac{7 \times 157}{4400} = .25 \text{ inch.}$$

Previous to 1892 the United States Treasury rules specified that the flues of a boiler should be made in lengths not exceeding three feet, in which case the following applies:

$$\frac{89{,}600\ T^2}{L\ D} = P$$

in which T=thickness, L=length of sections in feet, D=diameter in inches, and P=safe working pressure. Applying this to our 14-inch flue, our formula becomes

$$\frac{89600 \times .25 \times .25}{3 \times 14} = 133 \text{ pounds.}$$

Thus it will be seen that every part of this boiler is abundantly able to withstand the pressure to be put upon it, for the shell has a large margin, the braces are not overloaded, neither are they spaced so far apart as to introduce an element of danger, and even if flues are put into it instead of tubes, it is still safe under the designated pressure.

CHAPTER LIII.

CONCLUSION.

We have now arrived at the concluding chapter of this book, in which we have called attention to calculations that engineers are supposed to understand when they aspire to positions in places where a strict license law is in force. Occasionally we meet a man who claims that all of the duties of an engineer can be learned in a month. If we were to proceed to argue the case, we might be tempted to assert that possibly that portion of the business that he is familiar with could be learned within the specified time, but such arguments are not always profitable.

It is the author's experience that those engineers who study the most are the ones who are willing to allow that there is much more for them to learn, while oftentimes those who never spend any money or time in getting posted in the theoretical part of their business are the ones who claim to know it all. The author is willing to admit that the more a man studies steam engineering the more he will feel that there is more beyond, and that it is better further on, and this, too, after having spent several years in patient study, research and endeavors to improve every opportunity to gain knowledge on the subject.

It was our privilege some time ago to visit a panorama representing one of the greatest battlefields known to history. At the entrance we purchased a key explaining the entire view in detail,

but at first we were contented to gaze at the grand spectacle as a whole. Soon tiring of this we were about to depart, but opening the key and looking up the meaning of one part after another, and calling to mind the facts that they pictured to us, we soon found that, together with a friend, we were intensely interested, and when more than two hours had passed away, we had mastered only a portion of what was before us, but could not stay longer at that time. Shortly afterward, while conversing with a person who lived in that vicinity, we spoke of the grandeur of the sight, and of the wide extent of its teachings, but were coolly informed that it could all be understood in 15 minutes time.

So it is with some men in charge of steam plants, for they understand only what may be termed the superficial part of the business in which they are engaged, and consequently think that it can all be learned in a few weeks, but when they get a key to it (and we believe this to be a proper term, for it unlocks one of the doors to the storehouse of knowledge) and make use of it, they soon discover their mistake. It has been one of the objects of the author to make every subject treated of as plain as possible, and to introduce only those rules and formulas which are sanctioned by good authorities and that are based on common sense.

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TO

300 EXAMINATION

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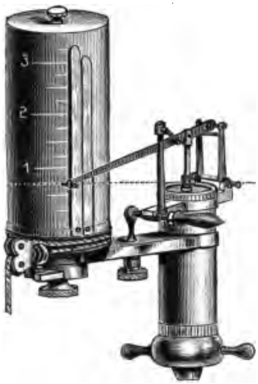

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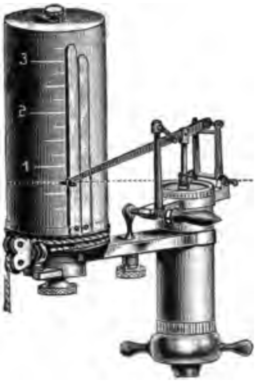

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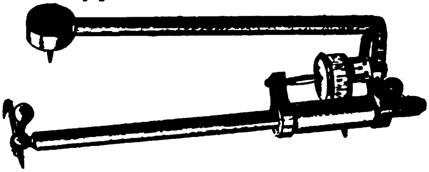

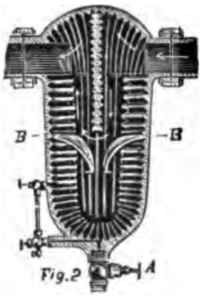

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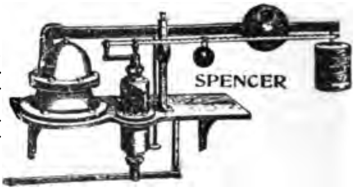

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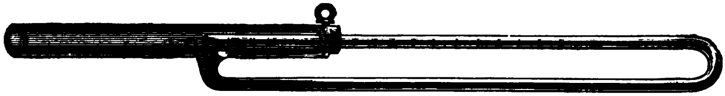

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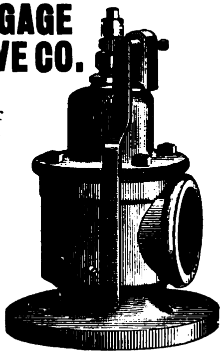

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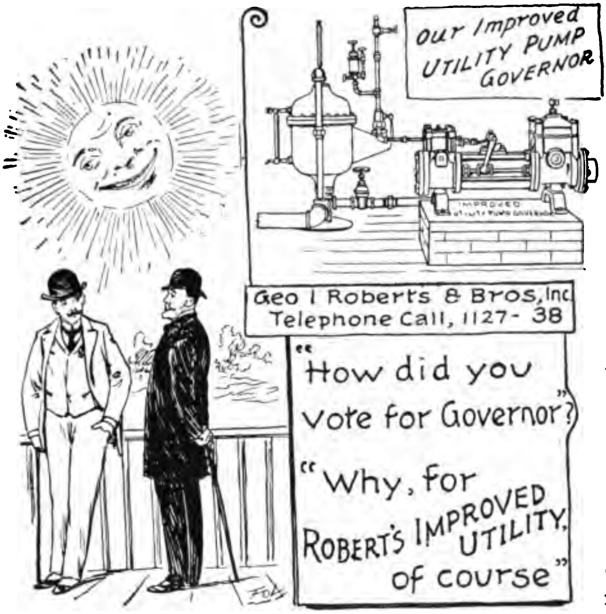

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